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Laboratory and Field Testing of Handpumps

Goh Sing Yau

Abstract An easily maintained handpump incorporating low-cost plastics (polyethylene and polyvinyl chloride) in the manufacture of the below-ground components has been developed for use in rural areas of the developing world. To ensure that it would function adequately under actual field conditions, laboratory and field testing protocols were developed by researchers at the University of Malaya, in collaboration with the Ministry of Health, Kuala Lumpur, Malaysia. This technical study contains a detailed description of these protocols. It includes a mathematical analysis of the reciprocating pump and a methodology for optimizing the design of the valve assemblies through computer simulation and selected laboratory testing. The methodology for field testing is described and standardized field survey forms are included. Two computer programs are also included for use on Apple II-compatible microcomputers. These permit the automatic acquisition and processing of data from an experimental rig.

Résumé On a mis au point, pour l'usage des habitants des régions rurales des pays en développement, une pompe à main d'entretien facile dont les pièces souterraines sont en matières plastiques bon marché (polyéthylène, chlorure de polyvinyle). Pour s'assurer que la pompe fonctionnerait convenablement dans les conditions réelles d'emploi, les chercheurs de l'Université de Malaya, en collaboration avec le ministère de la Santé de Kuala Lumpur, en Malaisie, ont élaboré un programme d'essai en laboratoire et sur le terrain. Cette étude technique présente la description détaillée de ce programme, incluant l'analyse mathématique de la pompe à mouvements alternatifs et une méthode d'amélioration de l'assemblage de la soupape grâce à la simulation informatique et à des essais spécieux en laboratoire. La méthodologie des essais sur le terrain est décrite et les formulaires normalisés d'enquête sur le terrain sont présentés, de même que deux logiciels utilisables sur des micro-ordinateurs compatibles avec Apple II. Ces logiciels permettent la collecte et le traitement automatique de données à partir d'un équipement expérimental.

Resumen Una bomba de fácil mantenimiento y que incorpora plásticos de bajo costo (polietileno y cloruro de polivinilo) en la manufactura de las partes subterráneas ha sido inventada para uso en las áreas del mundo en desarrollo. Para asegurar su funcionamiento adecuado en condiciones reales de campo, los investigadores de la Universidad Malaya desarrollaron protocolos de prueba en colaboración con el Ministerio de Salud de Malasia. Este estudio técnico contiene la descripción de tales protocolos. Incluye también un análisis matemático de la bomba y una metodología para optimizar el diseño de las partes de la válvula mediante simulación computarizada y pruebas de laboratorios selectas. Se describe la metodología para pruebas de campo y se incluyen formularios para las encuestas respectivas. También contiene este documento dos programas de computador para uso en microcomputadores Apple II compatibles. Todo esto permite la adquisición y procesamiento automáticos de datos del equipo experimental.

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FOREWORD

Because population growth has been rapid, material resources scarce, and expertise lacking, most developing countries have been unable to provide their rural populations with many of the basic services required to maintain their own physical and social well-being. Among the most important of these is the provision of safe adequate sources of drinking water.

In industrialized nations, an abundance of clean water is often taken for granted. In developing countries, however, water contaminated with disease-causing organisms may be all that is available. To focus attention on this growing problem, the International Water Supply and Sanitation Decade (1981-90) was launched. In response to the goals of the Decade, the International Development Research Centre (IDRC) has been providing support for research on the development of low-cost water-supply and sanitation technologies. As the provision of a clean water supply depends upon the existence of reliable pumping devices, IDRC has focused its efforts on the development of a simple low-cost handpump that uses readily available plastics such as polyvinyl chloride and polyethylene for the below-ground components. IDRC's objective was to encourage the development of a pump that could be manufactured locally and operated and maintained easily by largely uneducated rural populations. As with any technology, the technical and economic viability of the pump and its operational characteristics must be carefully assessed before it can be considered for actual use. This was accomplished by first simulating field conditions in the laboratory: the components were then modified according to the results of the tests and prototypes were built for testing under actual field conditions.

It is important that innovations in the development of any pumping device intended for local manufacture be durable and technically sound to function effec-

tively. To do this requires the development and implementation of rigorous testing procedures. This technical report describes the tests performed on the IDRC-University of Malaya (IDRC-UM) handpump, originally known as the Waterloo pump. The tests were developed by Professor Goh Sing Yau, principal investigator of both phases of the IDRC-supported project "Water Pumping Technology (Malaysia)."

IDRC's interest in publishing this document is to complement the Global Handpump Testing Programme of the United Nations Development Programme (UNDP) and the World Bank by sharing the knowledge and experience gained from several years of IDRC-supported research on the development of a low-cost handpump that can be operated and maintained at the village level. We hope that this document not only will contribute to the excellent work being carried out by the UNDP-World Bank's program but also will serve as a useful guide to those agencies, institutions, and industries now involved in handpump technology. This document, therefore, represents a contribution to the International Water Supply and Sanitation Decade and is aimed at benefiting the many millions of rural poor who still do not have access to one of our most precious resources -- clean water.

Donald S. Sharp
Associate Director (Water Supply and Sanitation)
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PREFACE

This book is intended to make available within one publication the results of a research project sponsored by the International Development Research Centre (IDRC) on the laboratory and field tests of polyvinyl chloride (PVC) handpumps that were carried out by the University of Malaya in collaboration with the Ministry of Health, Malaysia. Much of the material in this book has not been published previously although some has appeared separately in different publications. This book is aimed at handpump researchers as well as field workers.

Chapter 2 deals with the mathematical analysis of the reciprocating handpump and, with chapter 3 on laboratory testing, leads to a procedure for optimizing the detailed design of the valve assemblies of the handpump through computer simulation and selective laboratory testing. Chapter 4 describes the field tests. Thus, readers who are mainly interested in field testing may wish to miss chapters 2 and 3 and concentrate on chapter 4 and appendix C. Those who are already involved in or intend to begin laboratory testing may be more interested in chapters 2 and 3, and appendices A and B. Appendix A gives a listing of an interactive computer program based on the mathematical analysis of chapter 2 for the prediction of handpump performance. This allows a researcher to go straight into predicting handpump performance without having to repeat the work covered in chapter 2. Appendix B gives a listing of an interactive computer program for automatic acquisition and processing of data from the experimental rig described in chapter 3. Both these computer programs are written in Applesoft Basic for use in Apple II-compatible microcomputers. Thus the design and development of handpumps can be carried out using small and relatively inexpensive personal microcomputers.

My involvement in handpump research and development has been a great learning experience not only in the technical aspects but also through working with researchers from other disciplines both locally and abroad. This has been made possible by the generous support of the International Development Research Centre who funded the handpump project.

I would also like to thank several individuals with whom I have been involved: Tim Journey, Michael McGarry, and Marjorie Koblinsky for introducing me to the PVC handpump, and Donald Sharp and Lee Kam Wing for their support and continued interest in the handpump project.

Credit must also be given to the members of the project team (listed elsewhere in this publication) for their contributions and, last but not least, to the villagers in Kuala Pilah and Malim Nawar for their participation in the field tests.

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TERMINOLOGY AND DEFINITIONS

- A_p, A_R, A_o, A_h, A_a, A_v Cross-sectional area of pump cylinder, of rod, of orifice holes in piston, of space between valve flap and top face of piston, of space between edge of valve flap and pipe, of valve flap
- B_p, B_R Buoyancy force exerted by piston, by rod
- C_l Leakage constant
- d'_o, d_o, d_p Equivalent diameter of single large hole in piston, of diameter of each hole, of diameter of pump cylinder
- f Pipe friction factor
- F_u, F_D Friction force on piston, piston moving upward, moving downward
- F_{12}, F_{23}, F_{34}, F_{41} Force acting on piston rod in respective strokes
- g Gravitational acceleration
- h_1, h_2 Height of water column above piston, height of bottom face of piston from the bottom water level
- h_T Total height of lift of water
- k_1, k_2, k_c, k_T Fluid flow empirical constants
- ℓ Length of piston and foot valve
- L Distance moved by the piston

- L_r Length of connecting rod from flywheel to connecting pin at slide
- L_o Stroke length
- m Mass
- N Number of strokes applied per minute
- p, p_a, p_{w1}, p_{w2}, p_1, p_2 Pressures: general, atmospheric, above piston, below piston, at Station 1, at Station 2
- Δp, Δp_{12}, Δp_{23}, Δp_{34}, Δp_{41} Pressure drop across piston: general, and at respective strokes
- r Half stroke length
- R_e Reynold's number
- t Time
- t_{12}, t_{34} Time taken for valve flap to close at valve gap δ_{12}, δ_{34}
- U_o Velocity of flow in orifice of piston and foot valve
- $\dot{v}$ Leakage rate through piston rings
- V_L Leakage through piston rings per cycle
- W_p, W_R, W_{w1}, W_{w2} Weight of piston, of rod, of water column above piston, below piston
- x_{12}, x_{34} Displacement at which valve closes at respective stroke
- x Total valve delay
- y, $\dot{y}$, $\ddot{y}$ Distance, velocity, acceleration in y-direction
- z, $\dot{z}$, $\ddot{z}$ Distance, velocity, acceleration in z-direction
- α Wall effect correction factor

- δ_{12}, δ_{34} Valve gap for piston, foot valve
- μ Viscosity
- ρ, ρ_w, ρ_p Densities: general, of water, of plastic or brass
- ω_o Angular velocity

1. INTRODUCTION

In recognition of the urgent need to improve water-supply and sanitation facilities, especially for the rural population in developing countries, the United Nations declared the 1980s as the International Water Supply and Sanitation Decade with the objective of providing basic water-supply and sanitation facilities for all by 1990. It is unlikely that the targets of the decade for water supply can be met in developing countries solely through the use of large-scale piped water systems, not only because of the huge costs involved but also because of the shortage of local trained personnel to implement such schemes.

One of the simplest and least expensive methods for supplying reliable potable water is to install handpumps to extract groundwater wherever it is readily available in sufficient quantities. For many areas, handpumps can provide an interim solution but for the more remote and inaccessible parts of the world, the use of handpumps may be the only viable method to meet the basic water-supply needs of the rural population.

The hand-operated reciprocating pump for raising groundwater for human consumption has been in use since the 16th century with its basic design essentially unchanged. During the last 10 or 20 years, however, there has been a sudden surge of research and development activities on the reciprocating handpump (1-31) resulting in many new models and designs, mainly for use in developing countries. Some of these models incorporate parts made of indigenous materials such as wood (2, 10) and bamboo (3).

Since 1976, the International Development Research Centre (IDRC) has sponsored research and development work on a polyvinyl chloride (PVC) handpump design for use in developing countries (23). An essential

feature of the design is that it must be a village-level operated and maintained (VLOM) handpump. It must also be suitable for local fabrication in a developing country. This not only guarantees that there is technology transfer in terms of technical know-how but also ensures that spare parts are readily available locally.

The initial design work and laboratory tests on the IDRC handpump were carried out at the University of Waterloo in Canada (11). The piston and foot valve assemblies were subsequently tested at the Consumer Association's testing facilities in the United Kingdom (12, 13). In 1978, six projects were set up -- in Ethiopia, Malawi, Malaysia, the Philippines, Sri Lanka, and Thailand -- to carry out further laboratory tests and field trials of the common below-ground components (23): the above-ground components could differ from country to country and were designed with local fabrication in mind. This publication reports on laboratory and field testing techniques developed in the Malaysian project.

2. THE MATHEMATICAL MODEL

The fundamental principle of operation of the ideal basic cycle of the reciprocating water pump is fairly well understood. However, the analysis of the ideal cycle, such as that given by McJunkin (1) and Khurmi (32), does not account for the dynamic effects of fluid friction, pressure resistance due to fluid flow, valve delays, and leakage past piston seals, which, as is shown later, have a significant effect on the volumetric and mechanical efficiencies. These dynamic effects must be considered to allow the detailed design of the reciprocating pump to be optimized.

The Cycle

The reciprocating piston handpump considered here consists essentially of a riser pipe with two identical valves. The bottom valve, referred to here as the foot valve, is in a fixed position at the bottom of the riser pipe and is immersed in water. The upper valve, referred to here as the piston valve, is attached to a piston rod that moves the piston valve in a reciprocating motion a short distance above the foot valve.

The cycle of operation is illustrated in Fig. 1. State 1 shows the condition at the end of the previous downstroke. The piston valve is open and the foot valve closed. The upstroke consists of two characteristic portions. During the initial portion (upstroke 1 to 2), the piston valve is open and the piston rod experiences a relatively small force. The later portion (upstroke 2 to 3) starts immediately after the piston valve closes. This is effectively the "lifting" stroke as the piston rod takes the full pressure of the water column above the piston as well as the suction pressure below it. During this portion of the upstroke, water is drawn through the foot valve into the space between the piston and foot valve. Similarly, the downstroke

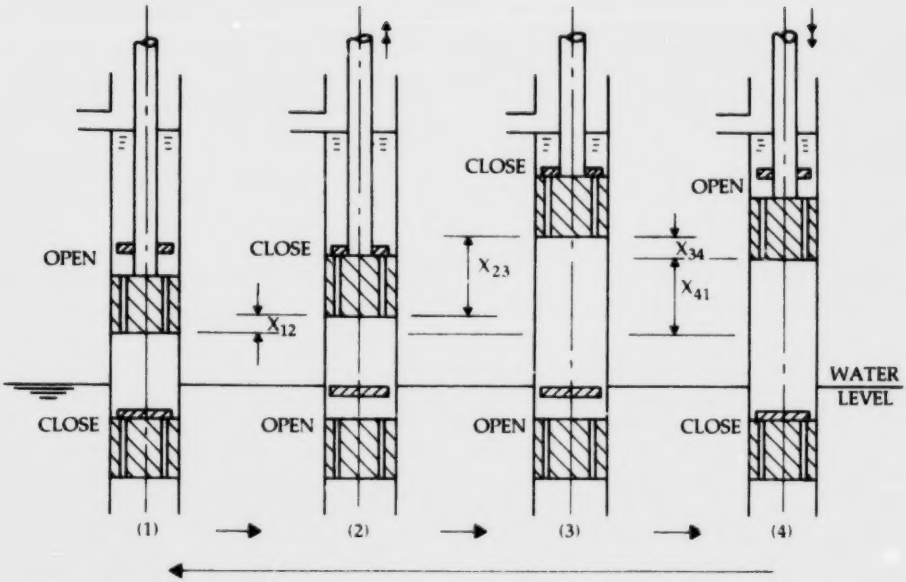

Fig. 1. Cycle of operation of reciprocating piston handpump. (From reference 23, page 40.)

consists of two characteristic portions. During the initial portion (downstroke 3 to 4), the piston rod experiences the full pressure of the water column above the piston as long as the downward acceleration of the piston is less than that of gravity and the foot valve is open. At state 4, the foot valve is closed and the later portion (downstroke 4 to 1) starts. The continuing downward movement of the piston keeps the piston valve open to allow water to rise above the piston until state 1 is reached again. During this latter portion of the downstroke, the piston rod experiences a relatively smaller force as it is relieved of the pressure of the water column above the piston by the opening of the piston valve.

Determination of Applied Force in the Piston Rod

Upstroke 1 to 2

The foot valve is closed and the piston valve open at the end of the downstroke. As the piston is lifted,

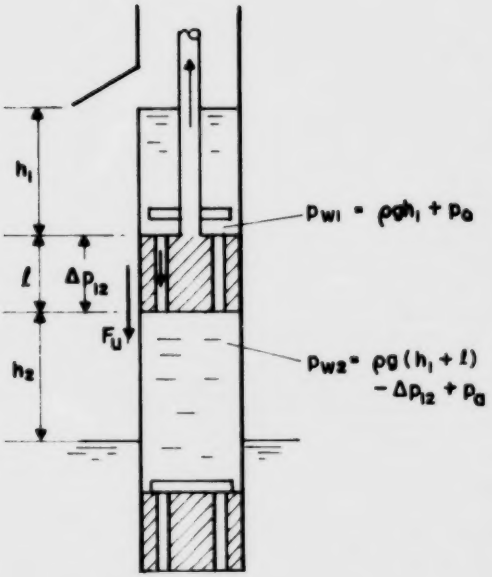

Fig. 2. Upstroke 1 to 2.

the forces acting on the piston are as indicated in Fig. 2.

The resultant force acting on the piston rod during this portion of the upstroke is given by

$$F_{12} = F_u + W_p + W_R + p_{w1}\,(A_p - A_R) + p_a A_R - p_{w2} A_p$$

$$= F_u + W_p + W_R + A_p\,(p_{w1} - p_{w2}) + A_R\,(p_a - p_{w1})$$

$$= F_u + W_p + W_R - A_p \rho g \ell - A_R \rho g h_1 + \Delta p_{12} . Ap$$

$$F_{12} = F_u + W_p + W_R - B_p - B_R + \Delta p_{12} . A_p \qquad [1]$$

where

F_u is the friction force on the piston when moving upward

W_p, W_R, are the weights of piston and piston rod respectively

P_{w1}, P_{w2}, P_a are pressures above piston, below piston, and the atmospheric pressure respectively

A_p, A_R are cross-sectional areas of piston and piston rod respectively

ρ is the density

g is the gravitational acceleration

ℓ is the length of piston and foot valve

h_1 is the height of water column above the piston

Δp_{12} is the pressure drop across the piston during upstroke 1 to 2

B_p, B_R are the buoyancy forces exerted by the piston and the piston rod respectively.

Upstroke 2 to 3

This is the portion of the upstroke after the piston valve closes and the piston "lifts" the water column trapped above it. The forces on the piston are as indicated in Fig. 3.

The resultant force acting on the piston rod is given by

$$F_{23} = F_u + W_p + W_R + p_{w1}(A_p - A_R) + p_a A_R - p_{w2} A_p$$

$$= F_u + W_p + W_R + A_p(p_{w1} - p_{w2}) + A_R(p_a - p_{w1})$$

$$= F_u + W_p + W_R + A_p \rho g(h_1 + h_2) + \Delta p_{23} A_p - A_R \rho g h_1$$

$$= F_u + W_p + W_R + \rho g h_1 (A_p - A_R) + A_R \rho g h_2 + \Delta p_{23} . A_p$$

$$F_{23} = F_u + W_p + W_R + W_{w1} + W_{w2} + \Delta p_{23} . A_p \qquad [2]$$

where

h_2 is the height of water column below piston

Δp_{23} is the pressure drop across the foot valve during upstroke 2 to 3.

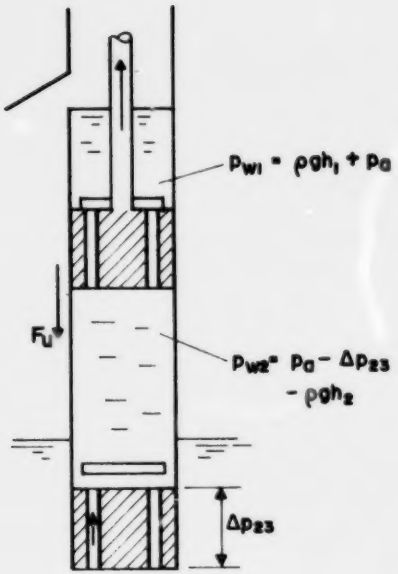

Fig. 3. Upstroke 2 to 3.

Downstroke 3 to 4

This is the initial portion of the downstroke when the piston valve is closed and the foot valve open. The forces acting on the piston are as indicated in Fig. 4.

The resultant force acting on the piston rod is given by

$$F_{34} = -F_D - A_p p_{w2} + p_{w1}(A_p - A_R) + A_R p_a + W_p + W_R$$

$$= -F_D + W_p + W_R + A_p(p_{w1} - p_{w2}) + A_R(p_a - p_{w1})$$

$$= -F_D + W_p + W_R + \rho g(h_1 + h_2) A_p - \rho g h_1 A_R - \Delta p_{34} A_p$$

$$= -F_D + W_p + W_R + \rho g h_1 (A_p - A_R) + \rho g h_2 A_p - \Delta p_{34} A_p$$

$$F_{34} = -F_D + W_p + W_R + W_{w1} + W_{w2} - \Delta p_{34} . A_p \qquad [3]$$

where

F_D is the friction force on the piston when moving downward

Δp_{34} is the pressure drop across the foot valve during downstroke 3 to 4.

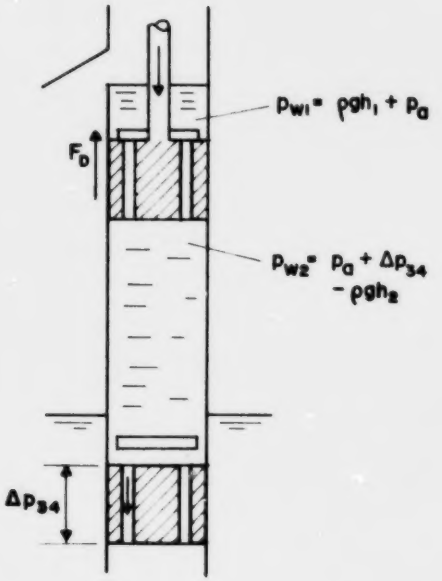

Fig. 4. Downstroke 3 to 4.

Downstroke 4 to 1

In the latter portion of the downstroke, the foot valve is closed and the water rises above the piston as the piston valve opens. The forces acting on the piston are as indicated in Fig. 5.

The resultant force acting on the piston rod is given by

$$F_{41} = -F_D + W_p + W_R + p_{w1}(A_p - A_R) + p_a A_R - p_{w2} A_p$$

$$= -F_D + W_p + W_R + A_p(p_{w1} - p_{w2}) + A_R(p_a - p_{w1})$$

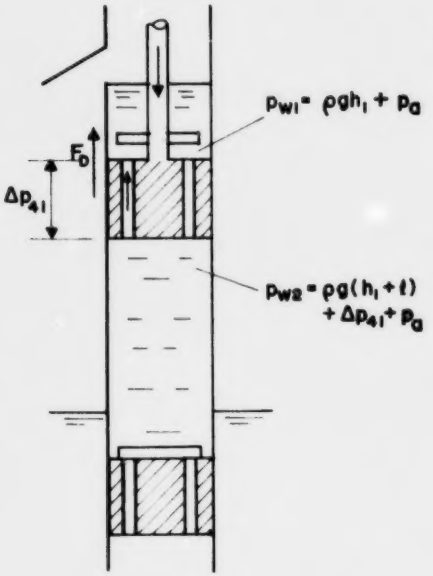

Fig. 5. Downstroke 4 to 1.

$$= -F_D + W_p + W_R - A_p(\rho g \ell + \Delta p_{41}) - A_R \rho g h_1$$

$$F_{41} = -F_D + W_p + W_R - B_p - B_R - \Delta p_{41} \cdot A_p \qquad [4]$$

where

Δp_{41} is the pressure drop across the piston during downstroke 4 to 1.

The Force-Displacement Diagram

Work Input

Using the four equations of the forces acting on the piston rod, a corresponding force-displacement diagram (Fig. 6) can be drawn for the cycle of operation of the reciprocating pump. The total work input over the cycle of operation is given by the area of the force-displacement loop. For the general case, the

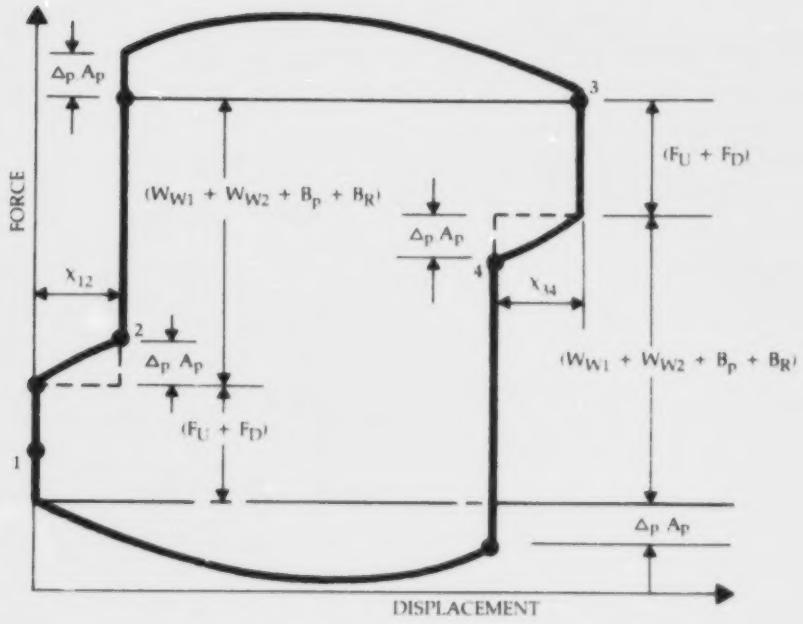

Fig. 6. Force-displacement diagram. (From reference 23, page 41.)

area of the force-displacement loop is given by expression

$$\begin{aligned}
\S F dL &= (W_{w1} + W_{w2} + B_p + B_R + F_u + F_D)(L_o - x_{12} - x_{34}) \\
&\quad + (F_u + F_D)(x_{12} + x_{34}) + \S \Delta p . A_p dL \\
&= [\rho g h_1 (A_p - A_R) + \rho g h_2 A_p + \rho g \ell A_p + \rho g h_1 A_R + F_u \\
&\quad + F_D](L_o - x_{12} - x_{34}) + (F_u + F_D)(x_{12} + x_{34}) \\
&\quad + \S \Delta p . A_p dL \\
&= A_p \rho g h_T L_o \, (1 - x/L_o) + F_T L_o + A_p \S \Delta p dL \qquad [5]
\end{aligned}$$

where

$h_T = h_1 + h_2$

$F_T = F_u + F_D$

$x = x_{12} + x_{34}$ (total valve delay)

L_o is the stroke length.

Volumetric Efficiency

The volumetric efficiency may be defined as

$$\eta_{vol} = \frac{\text{actual delivered volume flow per cycle}}{\text{volume displaced during suction stroke}}$$

$$= [(L_o - x_{12} - x_{34})A_p - V_L]/(L_o A_p)$$

$$= 1 - x/L_o - V_L/(L_o A_p) \qquad [6]$$

where

V_L is the total leakage through rings and valves.

The volumetric efficiency is, therefore, a measure of the wastage of possible volumetric output capacity. Valve delays and leakage through rings and valves decrease the volumetric efficiency.

Mechanical Efficiency

The mechanical efficiency may be defined as

$$\eta_{mech} = \frac{\text{"what we want"}}{\text{"what we have to do to get above"}}$$

$$= \frac{\text{water lifted to certain height}}{\text{work input}}$$

$$= \eta_{vol}\rho g L_o A_p h_T / \int F dL$$

$$= \frac{\rho g L_o A_p h_T [1 - x/L_o - V_L/(L_o A_p)]}{A_p \rho g h_T L_o (1 - x/L_o) + F_T L_o + A_p \int \Delta p dL}$$

$$= \frac{1 - V_L/[L_o A_p (1 - x/L_o)]}{1 + [F_T/A_p + (\int \Delta p dL)/L_o]/[\rho g h_T (1 - x/L_o)]} \qquad [7]$$

Where there are negligible leakage, friction, and pressure losses, i.e., $V_L = F_T = \Delta p = 0$, then, $\eta_{mech} = 100\%$. Note that x/L_o need not be zero. In other words, if there are negligible leakage, friction, and pressure losses, the valve delay has no effect on the mechanical efficiency. Thus, the mechanical efficiency may be considered as a measure of the wastage of effort as a result of leakage, friction, and pressure forces.

Further Physical Inputs into the Model

Pressure Drop across Piston and Foot Valve

The pressure drop across the piston and foot valve may be expressed as

$$\Delta p = \tfrac{1}{2} k_T \rho U_o^2 \qquad [8]$$

where

k_T is the overall pressure loss coefficient

U_o is the velocity in the orifice.

The pressure-loss integral in equation 7 may be evaluated to give

$$\int \Delta p \, dL = \int \tfrac{1}{2} k_T \rho U_o^2 \, dL$$

$$= \tfrac{1}{2} \rho k_T (A_p/A_o)^2 \int U_p^2 \, dL \qquad [9]$$

where

U_p is the average velocity in the riser pipe.

From a consideration of the geometry of the crank and connecting rod arrangement (Fig. 7), it may be shown that L is given by two expressions

for the upstroke

$$L = r[1 - \cos(\omega_o t)] - L_r \left[1 - \sqrt{1 - \left[\frac{r}{L_r} \sin(\omega_o t)\right]^2}\right] \qquad [10a]$$

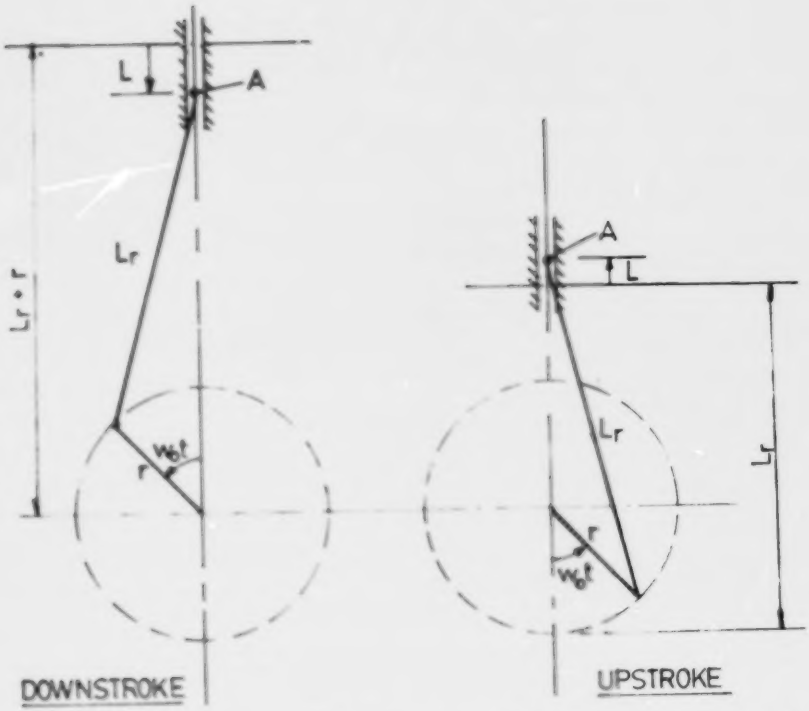

Fig. 7. Relative positions of crank during downstroke and upstroke (piston rod connected to pin A).

and for the downstroke

$$L = r[1 - \cos(\omega_o t)] + L_r[1 - \sqrt{1 - [\frac{r}{L_r} \sin(\omega_o t)]^2}] \quad [10b]$$

For a connecting rod much greater than the radius of the crank, i.e., $L_r >> r$, then the following apply

$$L = r[1 - \cos(\omega_o t)]$$

and

$$U_p = r\omega_o \sin(\omega_o t)$$

where

ω_o = is the angular velocity of the crank.

Substituting the above expressions in equation 9, we have

$$\S\Delta pdL = \rho k_T (A_p/A_o)^2 r^3 \omega_o^3 \int_o^{t_{\frac{1}{2}}} \sin^3(\omega_o t)dt$$

Putting $r = \frac{1}{2}L_o$ and evaluating the integral, we get

$$\S\Delta pdL = \frac{1}{6}\rho k_T (A_p/A_o)^2 L_o^3 \omega_o^2 \qquad [11]$$

For the present valve arrangement, the pressure drop across the piston and foot valve may be expressed as

$$\Delta p = \tfrac{1}{2}\rho U_o^2 [k_c + (1 - \frac{A_c}{A_p})^2 + \frac{f\ell}{d_o} + (\frac{A_o}{k_1 A_h})^2 + (\frac{A_o}{k_2 A_a})^2] \qquad [12]$$

where the first term in the parentheses refers to the pressure-loss coefficient for a sudden contraction; the second, a sudden expansion; the third, friction in the orifice; the fourth, radial flow between the valve flap and top face of piston and foot valve; and the fifth, flow between the valve edge and draw pipe.

Valve Closure Delays During Cycle of Operation

If the valve flap is light or the valve gap is large, there is a significant delay in the closure of the valves resulting in a drop in the volumetric efficiency of the pump. The magnitude of the valve delay may be determined as a function of weight of the valve flap, height of valve gap, and leakage rate past the piston rings.

Determination of "free-fall" time of valve flap

For a disc free-falling in a vertical cylindrical duct under its own weight, the drag force on the disc is given by

$$F_D = \alpha C_D A_v \tfrac{1}{2}\rho \dot{y}^2 \qquad [13]$$

where

$C_D = 1.3$ for turbulent flow (33)

d is the diameter of the disc

A_v is the area of the disc

α is the wall effect correction factor introduced to account for the increased drag caused by the confining wall of the duct

$\dot{y}$ is the relative velocity between the disc and the fluid.

The equation of motion for a disc free-falling in a water medium may be expressed as

$$m\ddot{y} + \tfrac{1}{2}\alpha C_D A_v \rho_w \dot{y}^2 - mg(1 - \rho_w/\rho_p) = 0 \qquad [14]$$

or

$$\ddot{y} + c\dot{y}^2 - g' = 0$$

where

$$c = \tfrac{1}{2}\alpha C_D A_v \rho_w / m$$

$$g' = g(1 - \rho_w/\rho_p)$$

Solving equation 14 for boundary conditions $t = 0$, $y = \dot{y} = 0$ gives

$$y = \frac{1}{c} \ln[\cosh(\sqrt{g'c}\, t)] \qquad [15]$$

Determination of x_{12}

The value x_{12} is the displacement of the valve flap as it closes during the upstroke.

For large ratios of orifice to valve area[1] The backflow of fluid through the holes in the piston where

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[1]Ratio of orifice to valve area is the ratio of the total area of the orifices in the valve to the cross-sectional area of the valve. Because the piston and foot valves are exactly the same in the IDRC-UM pump, the ratios are the same for each.

these are large in relation to the piston area is dependent only on the rate of rise of the piston. If the valve gap is δ_{12}, the time taken for the valve to close, t_{12}, may be obtained from the expression

$$\delta_{12} = y + z \qquad [16]$$

where

y refers to the "free-fall" distance derived earlier and z is the distance covered by the flap as a result of backflow through the piston holes.

For large ratios of orifice to valve area, the distance covered by the flap as a result of backflow through the piston holes is equal to the distance traveled by the piston.

The displacement at which the valve closes is, therefore, given by

$$x_{12} = z = L \qquad [17]$$

where

L may be obtained from equation 10a.

For small ratios of orifice to valve area The flow through the piston holes is dependent on the hole sizes where these are small in relation to the valve area. For very small holes, however, the backflow is negligible. The upward movement of the piston causes the foot valve to open almost immediately. As the second portion of the upstroke (2 to 3) starts immediately on the opening of the foot valve, the displacement x_{12} in this case is zero. The time taken for the valve flap to close may be obtained from the expression

$$\delta_{12} = y \qquad [18]$$

The value x_{34} is the displacement of the foot valve as it closes during the downstroke.

For large ratios of orifice to valve area It may be shown that, if the valve gap on the foot valve is δ_{34}, the time taken for the foot valve to close may be obtained from the expression

$$\delta_{34} = y + z + \dot{v}t_{34}/A_p \qquad [19]$$

where

$\dot{v}$ is the leakage rate through the piston rings.

The leakage through the piston rings is included here because the pressure difference across the piston is large during the initial portion of the downstroke (3 to 4).

The displacement of the foot valve as it closes is obtained from the relationship

$$x_{34} = z = L \qquad [20]$$

where

L may be obtained from equation 10b.

For small ratios of orifice to valve area The displacement and time taken for the foot valve to close when the orifices in the valve are small in proportion to the total valve area are given by

$$x_{34} = 0 \qquad [21]$$

$$\delta_{34} = y$$

Leakage Past the Piston Rings

Leakage past the piston rings during the suction stroke causes the volumetric and mechanical efficiencies of the pump to drop. The form of the leakage function may be determined as follows.

Consider the flow in the narrow passage between the outer surface of the piston ring and the inner wall of the pump cylinder to be that of Couette flow, we have from the Prandtl's Boundary Layer equation

$$\rho u \frac{du}{dy} + \rho v \frac{du}{dy} = \mu \frac{d^2u}{dy^2} - \frac{dp}{dx}$$

Using the Couette flow assumption where the convective terms are small compared with the shear stress and pressure gradient terms, we have

$$0 = \mu \frac{d^2y}{dy^2} - \frac{dp}{dx} \qquad [22]$$

Solving equation 22 for boundary conditions $y = 0$, $u = 0$ and $y = y_G$, $u = U_o$, we have

$$U = \frac{y}{y_G}\left[U_o - \frac{1}{2\mu}\frac{dp}{dx} y_G^2 \left(1 - \frac{y}{y_G}\right)\right] \qquad [23]$$

where

y_G is the gap between the outer surface of the piston ring and the pump cylinder

U_o is the relative velocity of the piston ring and the pump cylinder.

The total volume flow past the piston ring per cycle is, therefore, given by

$$V_L = \pi d_p \int_o^{y_G}\int_o^t U \, dy \, dt \qquad [24]$$

Substituting equation 23 and $U_o = r_o\omega_o \sin(\omega_o t)$ into equation 24 and integrating over the cycle, we have

$$V_L = \pi d_p y_G L_o \left(1 - \frac{5}{\mu}\frac{dp}{dx} y_G^2 \frac{1}{L_o N}\right) \qquad [25]$$

It may be shown in the present case, that the second term in the parentheses is much greater than unity. Therefore, equation 24 may be approximated by

$$V_L = \frac{-5\pi d_p}{\mu} y_G^3 \frac{dp}{dx} \frac{1}{N}$$

Putting $dp/dx = -\rho g h_T / t_r$ where t_r is the ring width, we have

$$V_L = \left(\frac{5\pi\rho g y_G^3}{\mu t_r}\right) \frac{d_p h_T}{N}$$

$$= C_1 (d_p h_T)/N \qquad [26]$$

Substituting equation 26 into equation 6, we have

$$\eta_{vol} = 1 - \frac{x}{L_o} - \left(C_1 \frac{d_p h_T}{A_p}\right) \frac{1}{L_o N} \qquad [27]$$

where

C_1 is a constant for a particular set of rings.

Any increase in spring stiffness will, however, decrease y_G and hence C_1. The value of C_1 may be determined from experimental data of volumetric efficiency for small ratios of x to L_o for different speeds of stroke application, N, and stroke length, L_o. A plot of the $(1 - x/L_o - \eta_{vol})$ against $1/L_oN$ gives a curve from which the gradient may be determined to give C_1.

Oscillations in the Piston Rod

We have previously shown that the force experienced by the piston rod changes depending on whether it is the initial or later portion of the upstroke or downstroke of the operating cycle. The loading and unloading of the piston rod at the instant of change of force cause oscillations to be induced in the piston rod.

These oscillations in the piston rod may be analyzed by assuming that the present arrangement is equivalent to an ideal spring-mass-dashpot systems (31) (Fig. 8). The actual oscillating mass is represented by a concentrated mass that is attached to a massless spring. A dashpot connected to the mass is provided to account for the viscous damping forces. The system is subjected to a simple harmonic disturbing force that acts through the support of the spring.

The equation of motion of mass m is given by

$$m\ddot{x} = k(y - x) - C_d\dot{x} - F$$

where the first term on the right-hand side of the equation is the spring force, the second term the viscous force, and the last term the external forces. Therefore,

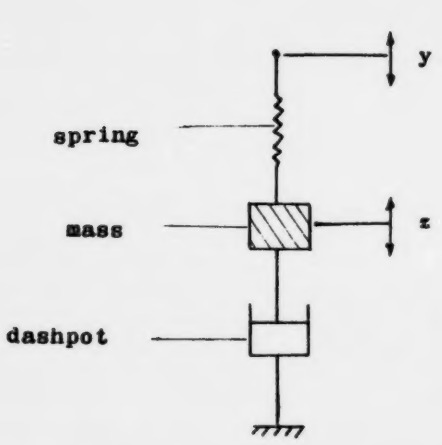

Fig. 8. Equivalent spring-mass-dashpot system.

$$m\ddot{x} - k(y - x) + C_d\dot{x} = -F$$

As $z = y - x$, we have

$$m(\ddot{y} - \ddot{z}) - kz + C_d(\dot{y} - \dot{z}) = -F$$

Therefore,

$$\ddot{z} + 2u\dot{z} + \omega_n^2 z = \ddot{y} + 2u\dot{y} + \frac{F}{m} \qquad [28]$$

where

$2u = C_d/m$

$\omega n^2 = k/m$

F is in the form $[F_i + 0.183\omega^2\cos(2\omega t)]$.

To determine the complementary function, consider the homogeneous equation

$$\ddot{z} + 2u\dot{z} + \omega_n^2 z = 0$$

The characteristic equation is given by

$$n^2 + 2un + \omega_n^2 = 0$$

Solving, we have

$$n = \frac{-2u \pm (4u^2 - 4\omega_n^2)}{2}$$

Assuming the oscillations are underdamped, we have

$$u < \omega_n$$

$$n = -u \pm i(\omega_n^2 - u^2)^{\frac{1}{2}}$$

$$= -u \pm iQ$$

$$z_{cf} = \exp(-ut)[A \sin(Qt) + B \cos(Qt)]$$

To determine the particular integral

$$\ddot{z} + 2u\dot{z} + \omega_n^2 z = r\omega^2 \cos(\omega t) + 2ur\omega \sin(\omega t) + \frac{F}{m} - \frac{0.183\omega^2 \cos(2\omega t)}{m} \qquad [29]$$

Assume a particular solution

$$z_{pi} = C \cos(\omega t) + D \sin(\omega t) + E + G \sin(2\omega t) + H \cos(2\omega t)$$

$$\dot{z}_{pi} = -C\omega \sin(\omega t) + D\omega \cos(\omega t) + 2\omega G \cos(2\omega t) - 2\omega H \sin(2\omega t)$$

$$\ddot{z}_{pi} = -C\omega^2 \cos(\omega t) - D\omega^2 \sin(\omega t) - 4\omega^2 G \sin(2\omega t) - 4\omega^2 H \cos(2\omega t)$$

Substituting z_{pi}, $\dot{z}_{pi}$, and $\ddot{z}_{pi}$ into equation 29 and equating the coefficients of $\ddot{z}$, $\dot{z}$, and z, the constants C, D, E, G, and H can be obtained.

The general solution is given by

$$z = z_{cf} + z_{pi}$$

$$= \exp(-ut)[A \sin(Qt) + B \cos(Qt)] + C \cos(\omega t) + D \sin(\omega t) + E + G \sin(2\omega t) + H \cos(2\omega t) \qquad [30]$$

where

$$u = C_d/2\,m$$

$$\omega_n^2 = k/m$$

$$Q = (\omega_n^2 - u^2)^{\frac{1}{2}}$$

$$C = \frac{r\omega^2(\omega_n^2 - \omega^2) - 4u^2\omega^2 r}{(\omega_n^2 - \omega^2)^2 + 4u^2\omega^2}$$

$$D = \frac{2ur\omega^3 + 2u\omega r(\omega_n^2 - \omega^2)}{(\omega_n^2 - \omega^2)^2 + 4u^2\omega^2}$$

$$E = F/m\omega_n^2$$

$$G = -0.732\omega^3 u/[m\{(\omega_n^2 - \omega^2)^2 + (4u\omega)^2\}]$$

$$H = G(\omega_n^2 - 4\omega^2)/4u\omega$$

and, depending on the stroke, constants A and B are given by

Stroke 1 to 2

$$B_1 = -C_1 - H_1 - \frac{0.183\omega^2}{m_1\omega_{n1}^2}$$

$$A_1 = \frac{u_1 B_1 - D_1\omega - 2\omega G_1}{Q_1}$$

Stroke 2 to 3

$$B_2 = z_{20} - C_2 - E_2 - H_2$$

$$A_2 = \frac{\dot{z}_{20} + u_2 B_2 - D_2\omega - 2\omega G_2}{Q_2}$$

where

$$z_{20} = e^{-u_1 t_1}[A_1\sin(Q_1 t_1) + B_1\cos(Q_1 t_1)] + C_1\cos(\omega t_1) + D_1\sin(\omega t_1) + E_1 + G_1\sin(2\omega t_1) + H_1\cos(2\omega t_1)$$

$$\dot{z}_{20} = r\omega\sin(\omega t_1)$$

Stroke 3 to 4

$$B_3 = z_{30} - C_3 - E_3 - H_3$$

$$A_3 = \frac{\dot{z}_{30} + u_3 B_3 - D_3\omega - 2\omega G_3}{Q_3}$$

where

$$z_{30} = e^{-u_2 t_2}[A_2 \sin(Q_2 t_2) + B_2 \cos(Q_2 t_2)] + C_2 \cos(\omega t_2) + D_2 \sin(\omega t_2) + E_2 + G_2 \sin(2\omega t_2) + H_2 \cos(2\omega t_2)$$

$$\dot{z}_{30} = -u_2 e^{-u_2 t_2}[A_2 \sin(Q_2 t_2) + B_2 \cos(Q_2 t_2)] + e^{-u_2 t_2}[A_2 Q_2 \cos(Q_2 t_2) - B_2 Q_2 \sin(Q_2 t_2)] - C_2 \omega \sin(\omega t_2) + D_2 \omega \cos(\omega t_2) + 2G_2 \cos(2\omega t_2) - 2H_2 \omega \sin(2\omega t_2)$$

Stroke 4 to 1

$$B_4 = z_{40} - C_4 - E_4 - H_4$$

$$A_4 = \frac{\dot{z}_{40} + u_4 B_4 - E_4\omega - 2\omega G_4}{Q_4}$$

where

$$z_{40} = e^{-u_3 t_3}[A_3 \sin(Q_3 t_3) + B_3 \cos(Q_3 t_3] + C_3 \cos(\omega t_3) + D_3 \sin(\omega t_3) + E_3 + G_3 \sin(2\omega t_3) + H_3 \cos(2\omega t_3)$$

$$\dot{z}_{40} = e^{-u_3 t_3}[A_3 Q_3 \cos(Q_3 t_3) - B_3 Q_3 \sin(Q_3 t_3)] - u_3 e^{-u_3 t_3}[A_3 \sin(Q_3 t_3) + B_3 \cos(Q_3 t_3)] - C_3 \omega \sin(\omega t_3) + D_3 \omega \cos(\omega t_3) + 2G_3 \omega \cos(2\omega t_3) - 2H_3 \omega \sin(2\omega t_3)$$

The force exerted on the piston rod is directly proportional to z, the solution of which is given in

equation 30. A mathematically more exact value of the total work input than that given in equation 5 may be obtained by integrating this oscillating force over the cycle. However, as the oscillations are lightly damped and energy dissipation is small, for practical purposes, we may ignore the oscillations and compute the total work input during the cycle from equation 5.

Computer Program for Predicting Pump Performance

A computer program for computing the volumetric and mechanical efficiencies based on the above mathematical analysis, but ignoring the effect of oscillations in the piston rod, is given in appendix A. The forces resulting from the acceleration and retardation of the water during the cycle (32) have also been ignored as they do not affect the total area enclosed by the force-displacement loop. The program is written in Applesoft Basic language suitable for use in Apple II-compatible microcomputers.

3. EXPERIMENTAL INVESTIGATION

Experimental Rig

The experimental arrangement for testing the handpump over a range of parameters consists essentially of a handpump assembly lifting water up to a maximum of 10 m from a central constant-level reservoir (Fig. 9). The assembly can also be converted to 3- or 6-m lifts of water when required. The water at the outlet of the handpump is returned to the central reservoir through a return pipe. The handpump is driven by a 746-W (1-horsepower) DC motor via a reduction gear and chain-drive assembly. The rotary motion of the flywheel is converted to a reciprocating vertical motion by a pin and slide. Mounting holes at various distances from the centre of the flywheel are provided to allow the stroke length to be changed. The speed of stroke application is changed by varying the speed of the DC motor. This is achieved by varying the input voltage of the armature coil while keeping a full 240 V across the field coil of the DC motor.

The force in the piston rod is measured by four strain gauges on a proof ring attached to the top end of the piston rod and the piston displacement is measured by a displacement transducer (Figs. 10 and 11). Both the force and displacement signals go to a dynamic strain meter via separate bridges and then into a Tektronix Digital Storage Oscilloscope 5223.[1] The signals are digitized and transmitted to an Apple II-compatible microcomputer via a general purpose interphase bus (GPIB). An interactive computer program, which is listed in appendix B, was developed to retrieve data from the strain gauges and displacement transducer; to

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[1]Mention of proprietary names does not constitute endorsement of the product and is given only for information.

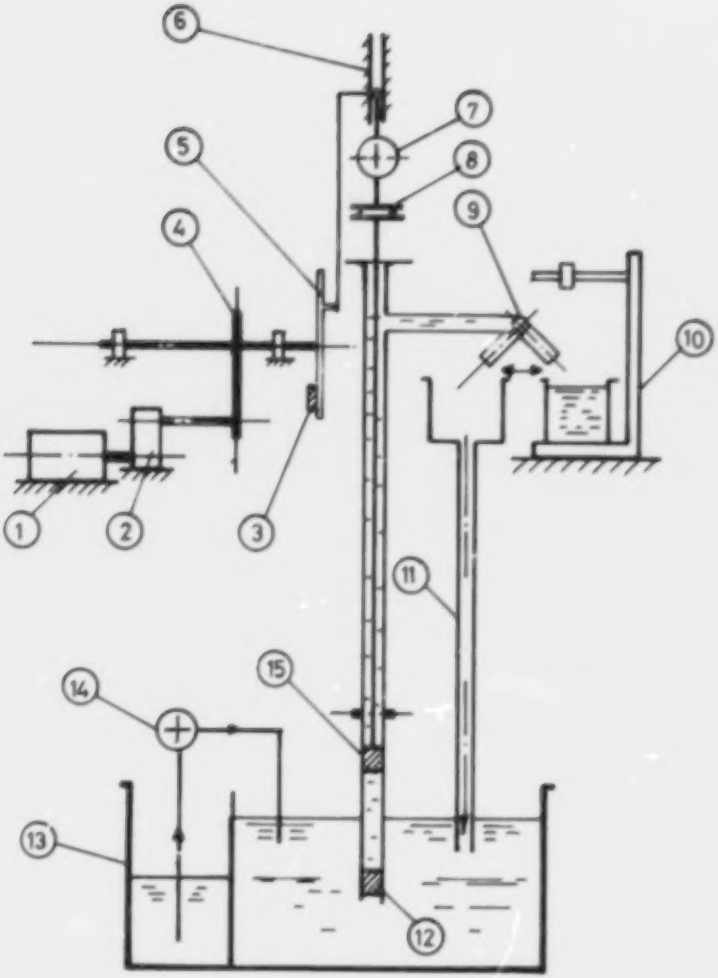

Fig. 9. Schematic of handpump testing assembly: (1) DC motor, (2) gear box, (3) counterweight, (4) chain drive, (5) flywheel, (6) pin and slide, (7) proof ring, (8) mechanical instrument, (9) swivel outlet, (10) weighing machine, (11) return pipe, (12) foot valve, (13) reservoir, (14) recycling pump, and (15) piston.

input data on calibration, experimental conditions, and work output; to determine the period of the cycle of operation; and to integrate the force-displacement diagram through the use of a trapezoidal rule formula.

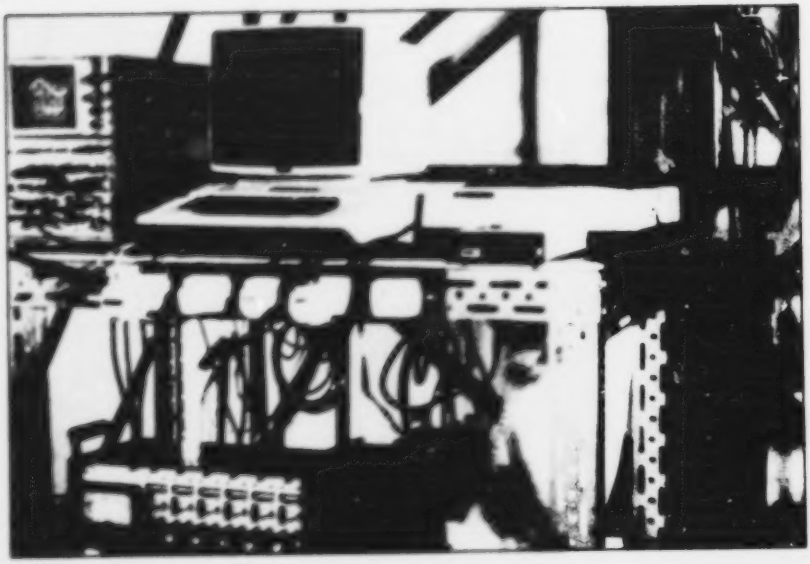

Fig. 10. Instrumentation for measuring pump performance.

Finally, the volumetric and mechanical efficiencies are computed. Hard copies of the results can be obtained through a printer to give printouts of curves of force versus displacement and plots of volumetric or mechanical efficiency versus speed of stroke application over several stroke lengths.

Comparison of Theoretical Predictions of Mathematical Model with Experimental Data

Oscillations Generated in the Piston Rod

Experimental and predictive results of the extension-time plot of the oscillations in the piston rod generally agree well (Fig. 12).

Upstroke 1 to 2

During the first portion of the upstroke, the piston valve is open and the piston rod experiences a relatively small force, which is confirmed by the small extension in the experimental data for this portion of

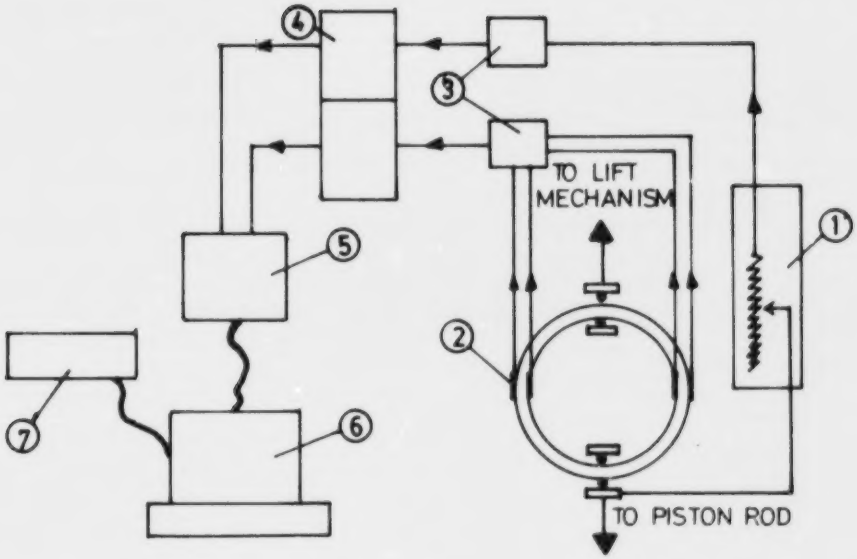

Fig. 11. Instrumentation for measurement of mechanical efficiency: (1) displacement transducer, (2) strain gauges on proof ring, (3) straing gauge bridge, (4) dynamic strain meter, (5) digital storage oscilloscope, (6) microcomputer, and (7) printer.

the cycle. The effective oscillating mass is the sum of the masses of the piston and piston rod and a fraction K of the mass of the water column. It was found that the value of K varies with the pumping speed as well as with the water head.

Upstroke 2 to 3

Upstroke 2 to 3 starts immediately after the piston valve closes and the piston rod takes up the full load of the water column. The piston rod is treated analytically as being acted on by an impact load that sets up damped oscillations in itself. The effective oscillating mass is the sum of the masses of piston and piston rod, and the mass of water column above it.

Except for the first half cycle, the frequency of the oscillations is constant over this portion of the upstroke and is independent of the pumping speed (Fig. 12).

Downstroke 3 to 4

Stroke 3 to 4 is the initial portion of the downstroke. The piston valve is closed and the piston rod experiences the full load of the water column above the piston. There is a sight drop in extension caused by the change in direction of the frictional force when the stroke is reversed. This portion of the stroke appears as a slight discontinuity after stroke 2 to 3.

Downstroke 4 to 1

The latter portion of the downstroke starts immediately after the foot valve closes. The piston rod is relieved of the load of the water column and the sudden unloading sets up damped oscillation in the piston rod. The frequency of oscillation is higher than that of upstroke 2 to 3 because the effective oscillating mass is smaller.

General Characteristics of the Force-Displacement Diagram

Figure 13 is an example of the trace of the force-displacement diagram for a trial with large valve-closure delays (x_{12} and x_{34}) caused by the use of large valve gaps and light valve flaps. The ratios of orifice to valve area are relatively large so that the pressure drops across the piston and foot valve are small. The figure shows that the sudden take-up or release of load set up damped oscillations in the piston rod as described in the previous section.

When the valve gaps are reduced (Fig. 14), the trace of the force-displacement diagram shows that the valve closure delays are negligible. On the other hand, the small valve gaps cause the pressure drops across the piston and foot valve to increase, resulting in a large increase in work output.

As the oscillations in the actual force-displacement diagram are lightly damped, energy dissipation resulting from these oscillations is relatively small and, therefore, may be neglected. This is confirmed by close agreement between the area of the force-displacement diagram actually measured and that predicted using equation 5.

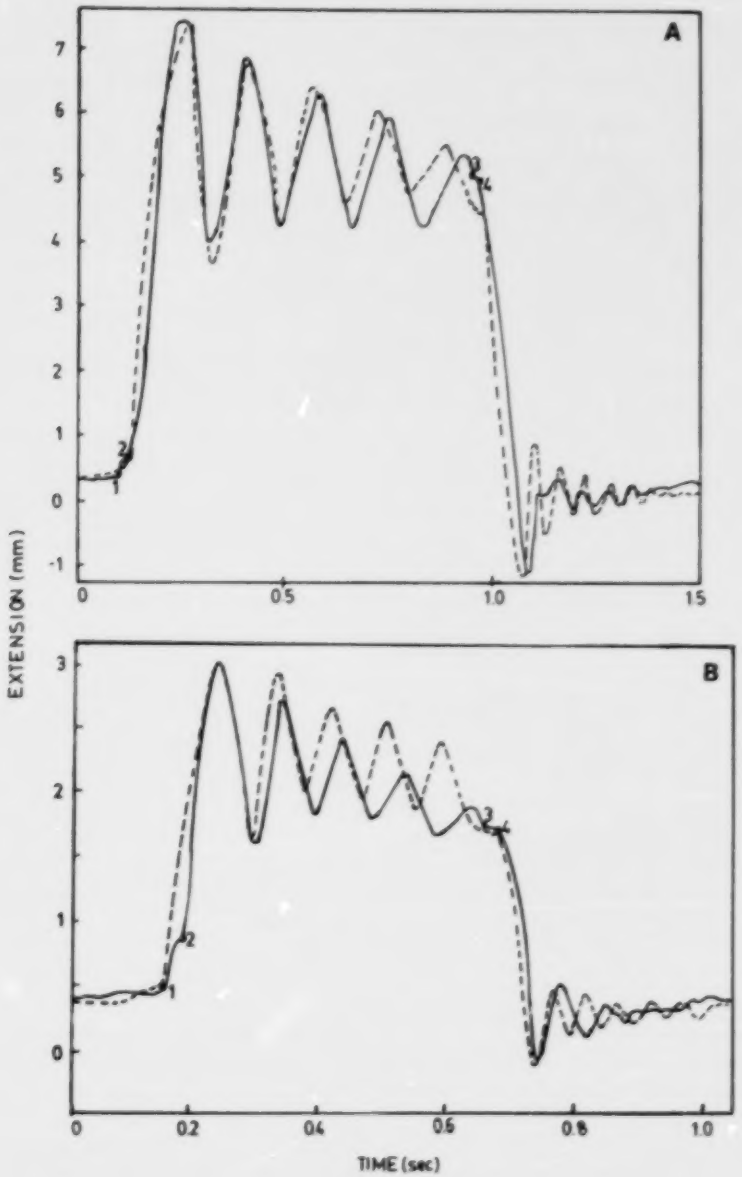

Fig. 12. Oscillations generated in the piston rod: **(A)** for pumping speed of 60 strokes/min at 3-m water head, and **(B)** for pumping speed of 40 strokes/min at 6-m water head. (Solid lines are experimental data and broken lines are predictions.)

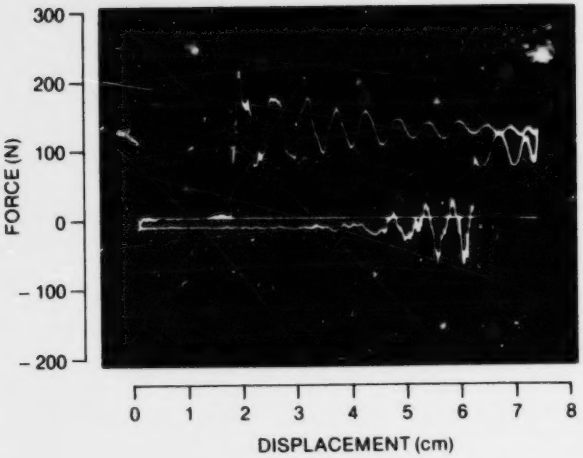

Fig. 13. Force-displacement diagram with large valve-closure delays.

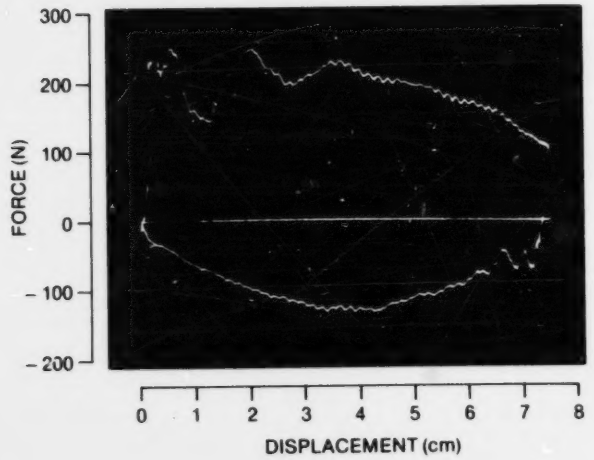

Fig. 14. Force-displacement diagram with negligible valve-closure delays but with large pressure drops across foot valve and piston.

Determination of Leakage Function

It may be observed from equation 26 that the leakage "constant" C_l is equal to $5\pi\rho\ g y_G^3/(\mu t_r)$ where

ρ, g, and μ are constants at a particular temperature, y_G is the gap between the ring and the pump cylinder, and t_r is the longitudinal thickness of the piston rings.

If the outward spring force exerted by the piston ring against the cylinder is increased, y_G and C_1 decrease resulting in increased sealing efficiency.

As polyethylene "creeps," it is important when testing for C_1 to ensure that the polyethylene piston rings have settled into the cylinder and that the outward spring tension of the piston ring is relatively constant, otherwise the sealing efficiency will vary with time and the results will be inconsistent, especially when the results are obtained over several days.

Figure 15 is an example of a plot of $(1 - x/L_o - \eta_{vol})$ against $1/L_oN$ for a set of piston rings. The solid line gives a linear relationship from which the leakage function may be determined. In this case, the leakage function (in m^3/cycle) is given by

$$V_L = 3.08 \times 10^{-3} \frac{d_p h_T}{N} \qquad [31]$$

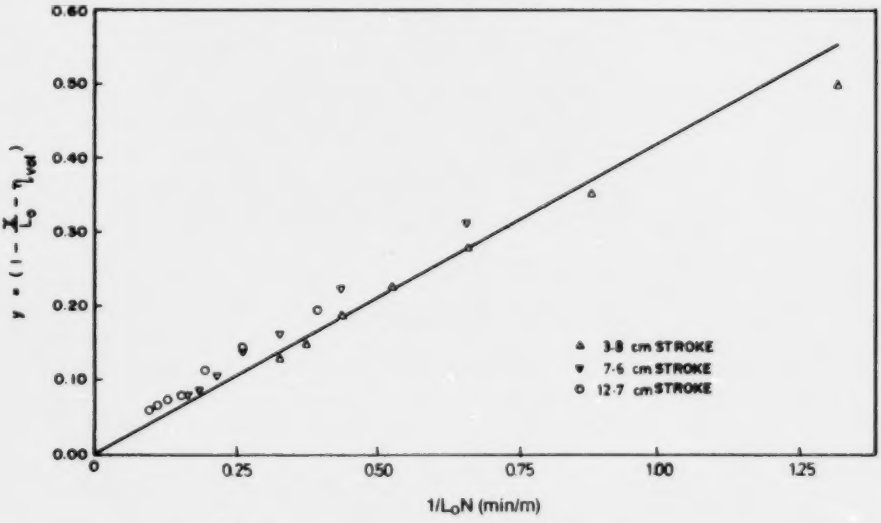

Fig. 15. Determination of leakage function for piston ring set number 1.

Friction Force on the Rings

To compute the mechanical efficiency from equation 7, it is necessary to know the value of F_T, the sum of the upward and downward frictional forces between the piston and the wall of the pipe. Figure 13 is an example of an oscilloscope trace of the force-displacement diagram where the valve delay is large. It may be observed that F_T is relatively constant with piston speed and equal to 15.6 N.

Variation of Volumetric Efficiency with x/L_o Ratio

Volumetric efficiency varies with varying x/L_o ratio (Fig. 16). The points are experimental data for two different stroke lengths and three different valve-flap weights and agree well with the theoretical predictions (solid lines) based on equation 6.

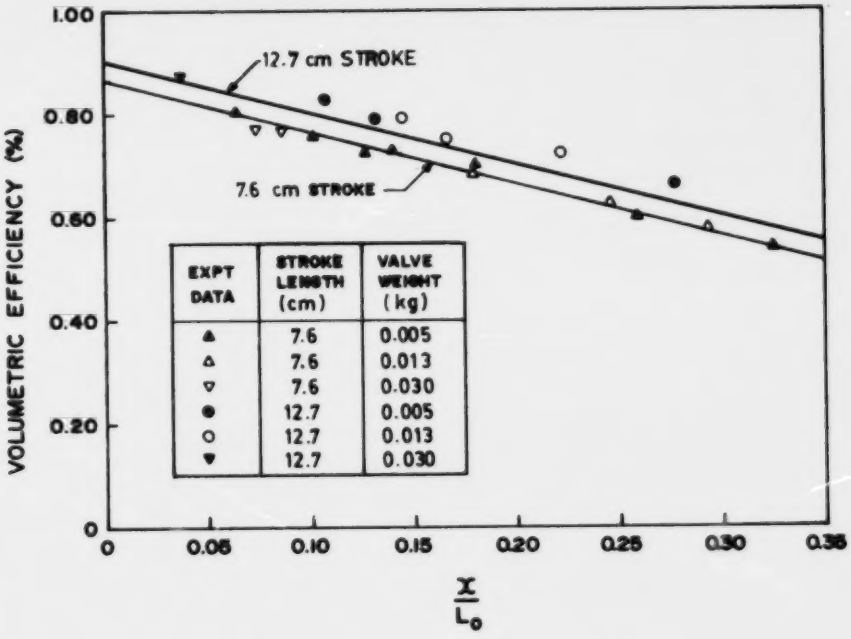

Fig. 16. Variation of volumetric efficiency with x/L_o ratio (solid lines are predictions).

Variation of x/L_o Ratio with Valve Gap and Valve-flap Weight

The valve closure delay ratio, x/L_o, varies both with valve gap and valve weight (Fig. 17). The value of the wall correction factor, which was introduced into equation 13 to account for the increase in drag caused by the presence of the confining wall of the duct, was found to be equal to 2 in order to procure reasonable agreement between the predicted lines and the experimental data.

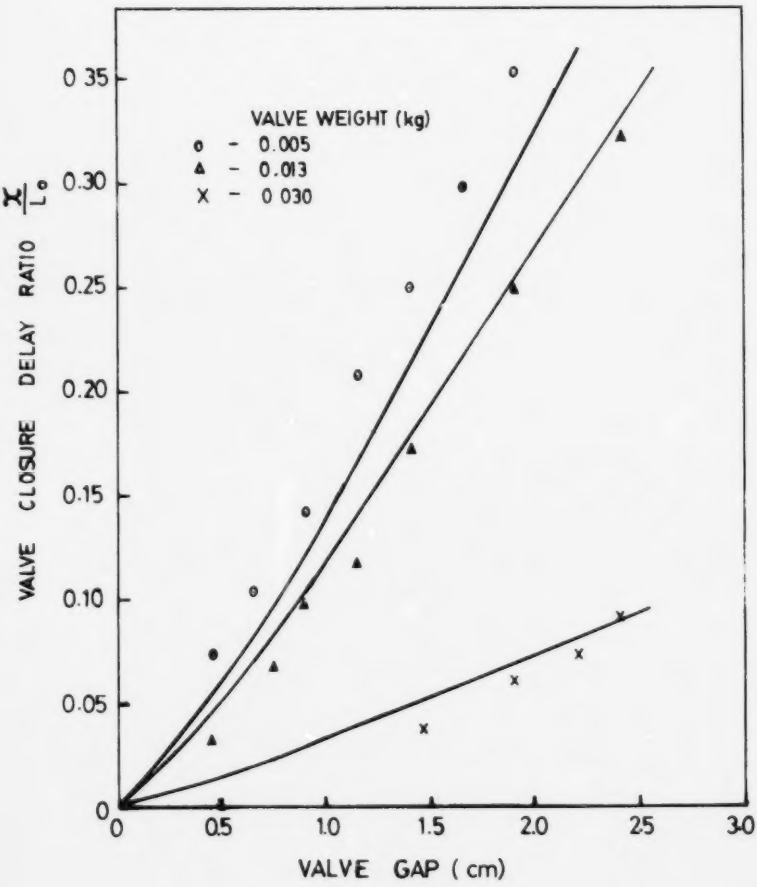

Fig. 17. Variation of x/L_o ratio with valve gap and weight of valve flap (solid lines are predictions).

Variation of Volumetric and Mechanical Efficiencies

Effect of speed of stroke application and stroke length

The volumetric efficiency increases with increasing speeds of stroke application and with increasing stroke length (Fig. 18). As the valve gap for this set of experiments was 2.87 mm, the valve closure delay ratio, x/L_o, was small for all cases. Examination of equations 6 and 31 shows that the drop in volumetric efficiency at low speeds of stroke application and shorter strokes is caused by increased leakage past the rings at low piston speeds. This close agreement between the experimental data and theoretical predictions confirms the correct choice of the leakage function as described earlier.

The corresponding mechanical efficiency varies less uniformly with speed of stroke application (Fig. 19). It is low for low speed of stroke application for the 3.8-cm

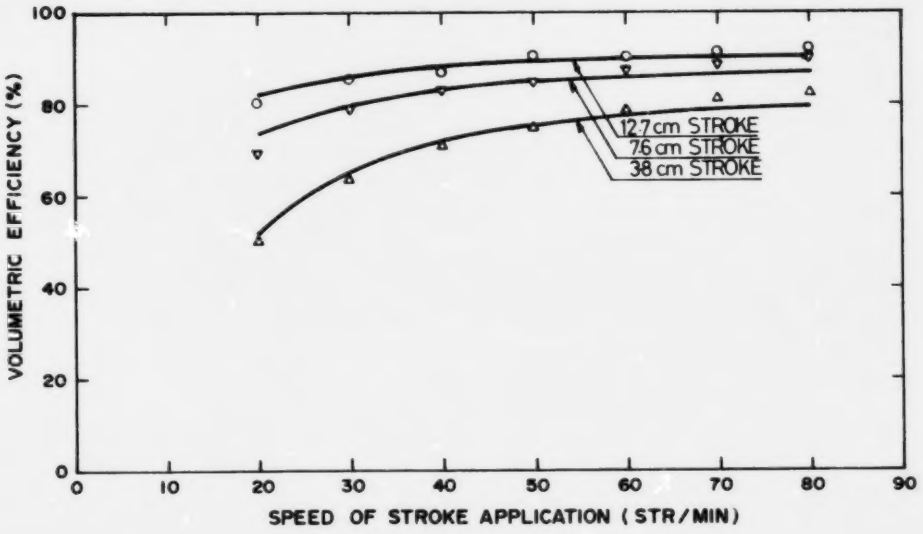

Fig. 18. Variation of volumetric efficiency with speed of stroke application for three stroke lengths (points are experimental data and solid lines are predictions based on equations 6 and 31).

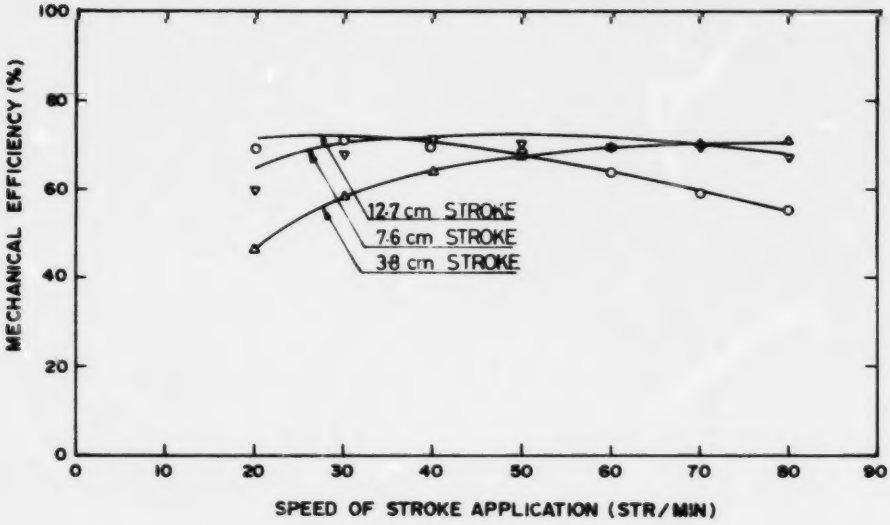

Fig. 19. Variation of mechanical efficiency with speed of stroke application for three stroke lengths (points are experimental data and solid lines are predictions based on equation 7).

stroke length as is expected because leakage past the piston rings is high at low piston speeds. It is also low for high speeds of stroke application for the 12.7-cm stroke. In this case, the leakage past the piston rings is much reduced because of the high piston speeds, but the high piston speeds have also resulted in a large increase in the pressure drop across the piston and foot valve causing a net decrease in mechanical efficiency. The mechanical efficiency remains fairly constant over a large range of speeds of stroke application for the 7.6-cm stroke length.

Effect of valve gap for different weights of valve flap

The volumetric efficiency decreases with increasing valve gaps and decreasing weights of the valve flap (Fig. 20) as is expected because increasing valve gaps and decreasing weights of the valve flap increase the valve closure delay ratio, x/L_o.

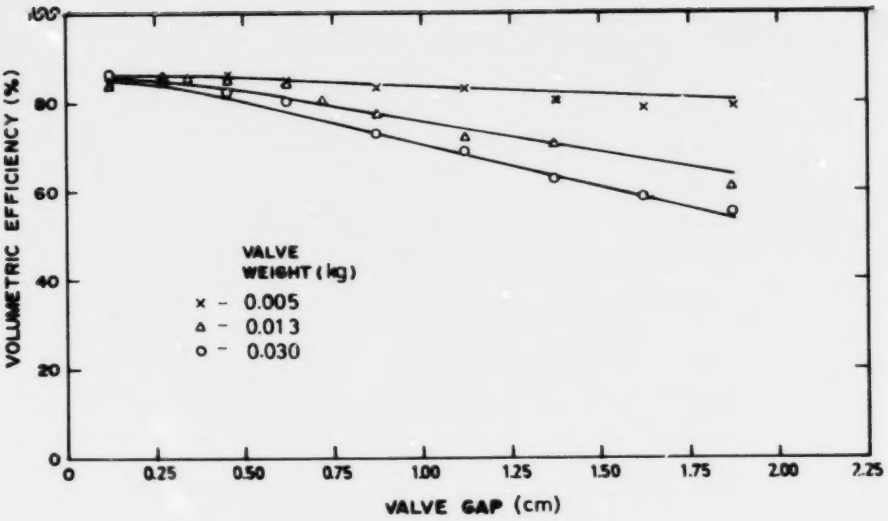

Fig. 20. Variation of volumetric efficiency with valve gap for three different valve weights (points are experimental data and solid lines are predictions).

The corresponding mechanical efficiency increases slightly with decreasing valve gap until the valve gap is of the order of 0.25 cm (Fig. 21). A further decrease in valve gap causes a large increase in pressure drop across the piston and foot valve resulting in a drop in mechanical efficiency.

Effect of ratio of orifice to valve area

The volumetric efficiency is not significantly affected by changes of ratio of orifice to valve area from 7.3 to 24.0% (Fig. 22) because volumetric efficiency is a function only of the valve delay ratio and the leakage.

The corresponding mechanical efficiencies (Fig. 23) show that a decrease in ratio of orifice to valve area decreases the mechanical efficiency. This is as expected because small ratios of orifice to valve area increase the pressure drop across the piston and foot valve resulting in increased work input and hence decreased mechanical efficiency.

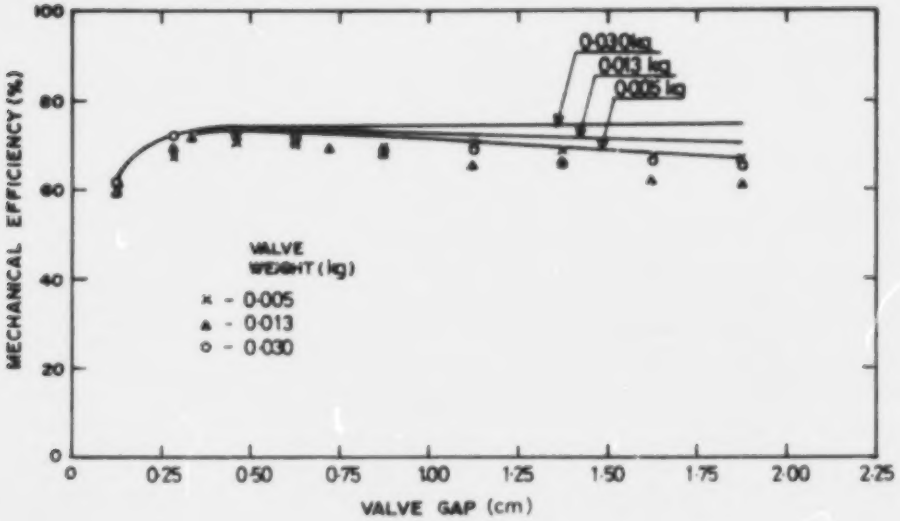

Fig. 21. Variation of mechanical efficiency with valve gap for three different valve weights (points are experimental data and solid lines are predictions).

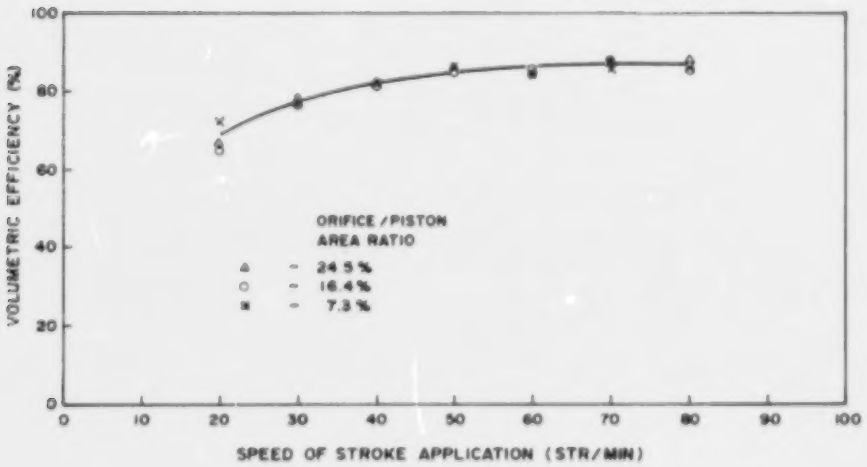

Fig. 22. Variation of volumetric efficiency with speed of stroke application for different ratios of orifice to valve area (points are experimental data and solid lines are predictions).

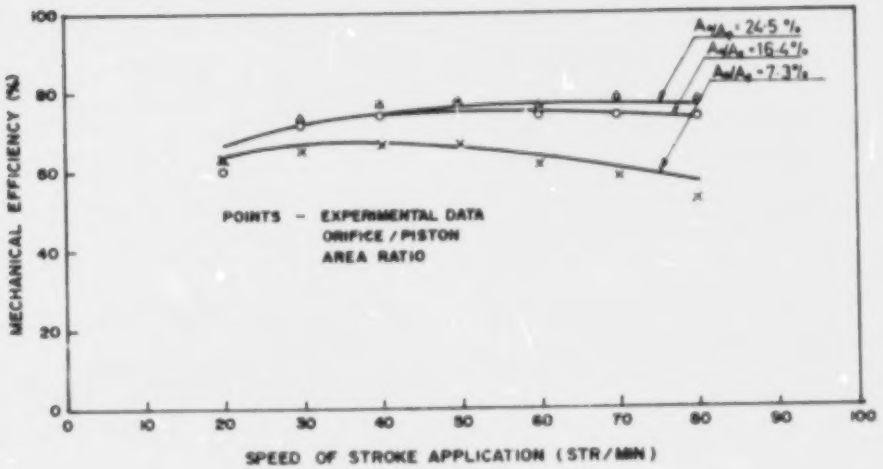

Fig. 23. Variation of mechanical efficiency with speed of stroke application for different ratios of orifice to valve area (points are experimental data and solid lines are predictions).

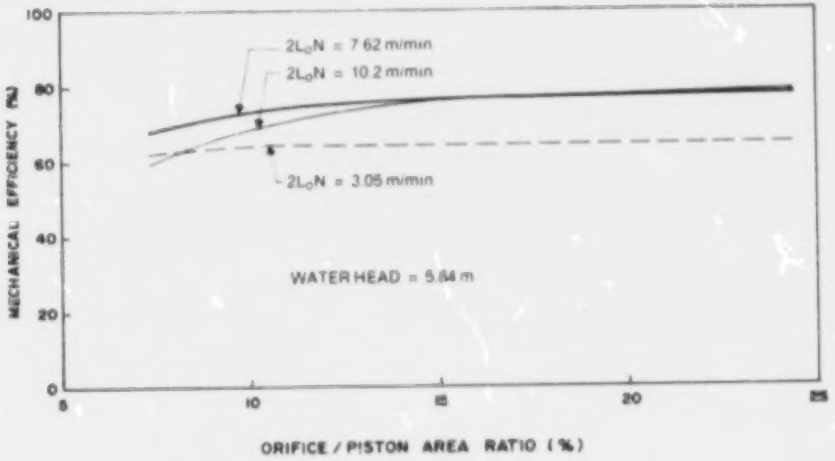

Fig. 24. Variation of mechanical efficiency with ratio of orifice to valve area.

By substituting equations 11 and 26 into equation 7 and putting $S = 2L_oN$, where S is the mean piston speed, it may be shown that, for cases where x/L_o is small, the mechanical efficiency is a function of S.

Figure 24 is an example of predictions of the mechanical efficiency against the ratio of orifice to valve area for three piston speeds. For a piston speed of 3.05 m/min, the mechanical efficiency is almost independent of the ratio because, at the low velocities of water flow through the orifices in the piston and foot valve, the pressure losses are small. At higher piston speeds, 7.62 and 10.2 m/min, the mechanical efficiency curves dip when the area ratio is less than 15%. It may be noted that the mechanical efficiency is generally higher at a higher piston speed. This is to be expected as the sealing efficiency of the piston rings is higher at a higher piston speed. The optimum ratio of orifice to valve area appears to be 15% (Fig. 24) and there is little added improvement in mechanical efficiency with a further increase in area ratio.

Effect of water head

For the following series of tests, a new set of piston rings was used. Figure 25 is an example of a plot of $(1 - x/L_o - \eta_{vol})$ against $1/L_oN$ for a water head of 8.25 m. The solid line gives a linear relationship from which the leakage function may be determined. In this case, the leakage function (in m^3/cycle) is given by

$$V_L = 1.28 \times 10^{-3} \frac{d^2 h_T}{N} \qquad [32]$$

Theoretical predictions of volumetric efficiency (using equations 6 and 32) are in fair agreement with experimental data for three different levels of water head and three different stroke lengths (Fig. 26). The experimental results show that the volumetric efficiency decreases with water head as is expected because the leakage past the piston rings per cycle of pump operation is directly proportional to the water head.

The corresponding mechanical efficiencies (Fig. 27) show that, at low piston speeds, an increase in water head decreases the mechanical efficiency. At high piston speeds, however, an increase in water head increases the mechanical efficiency. This behaviour is in agreement with equation 7. At low piston speeds, the leakage is large and the pressure loss term is small. An increase in water head therefore decreases the

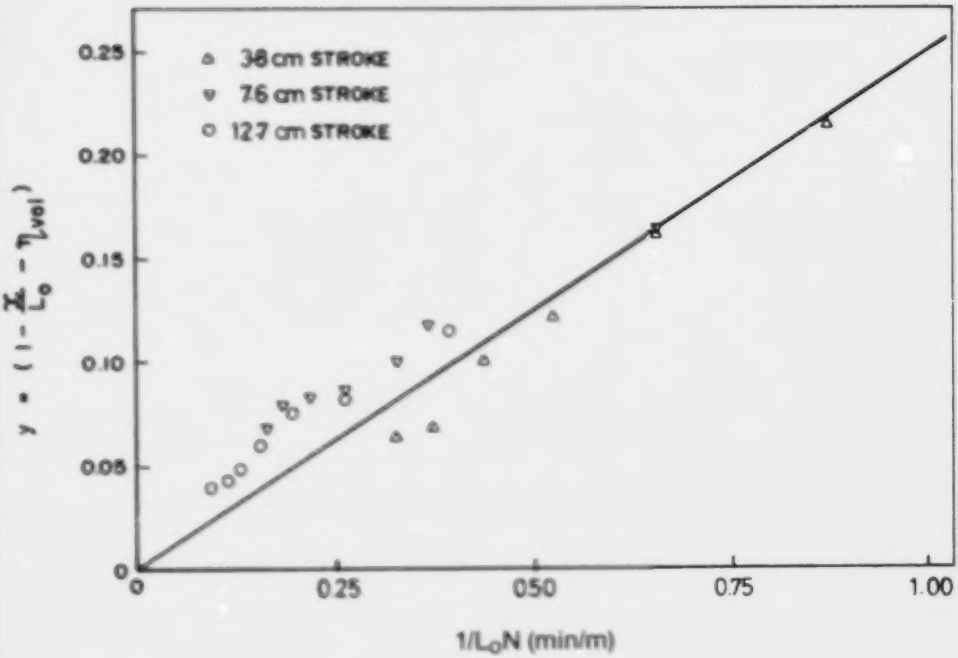

Fig. 25. Determination of leakage function for piston ring set number 2.

numerator more than the denominator on the left-hand side of equation 7 resulting in a decrease in mechanical efficiency. On the other hand, at high piston speeds, the leakage is small and the pressure loss term is large. An increase in water head, therefore, decreases the denominator more than the numerator resulting in an increase in mechanical efficiency.

Effect of conical entry in piston and foot valve

Neither the volumetric nor mechanical efficiencies change significantly with a conical rather than an abrupt entry to both the piston and foot valve (Fig. 28) as is confirmed by the theoretical predictions. For a conical entry, the pressure loss coefficient for the sudden contraction, k_c in equation 12, is reduced. The theoretical predictions for a perfectly streamlined entry ($k_c = 0$) shows that even at the highest piston speeds (12.7-cm stroke and 80 strokes/min), the

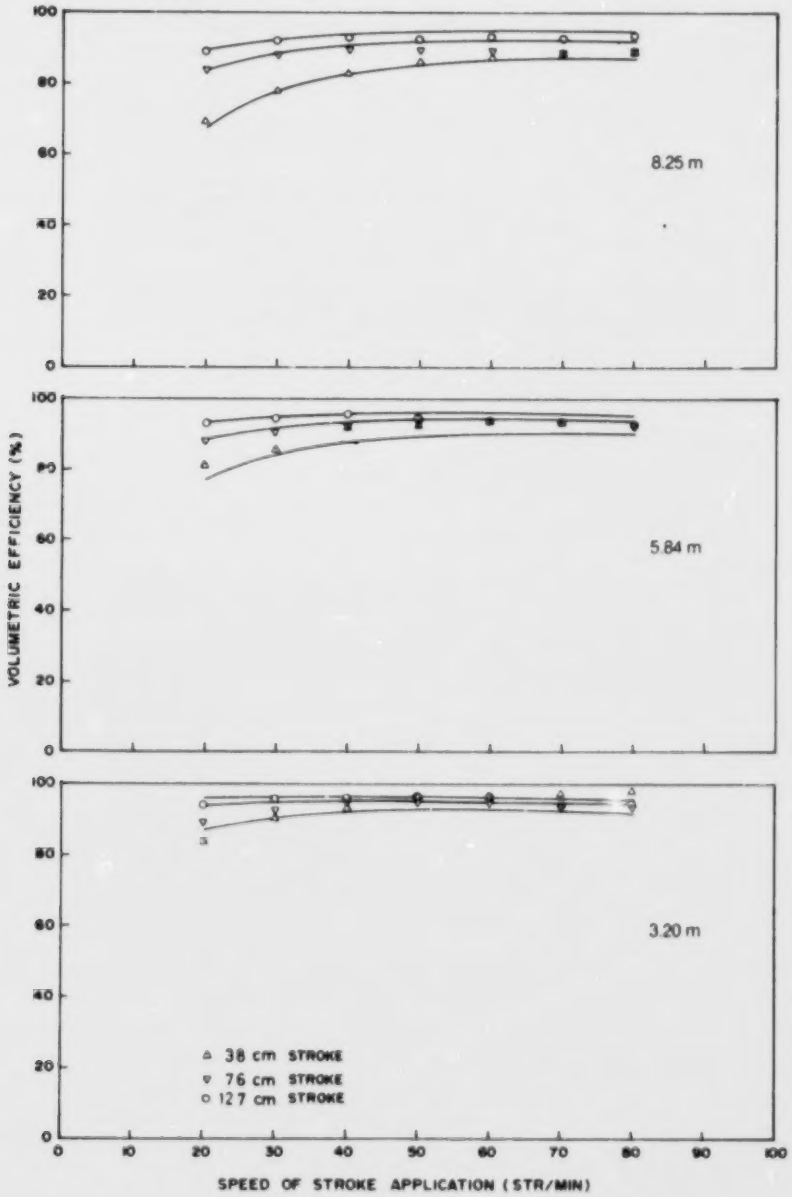

Fig. 26. Variation of volumetric efficiency with speed of stroke application at 8.25, 5.84, and 3.20 m water head (points are experimental data and solid lines are predictions).

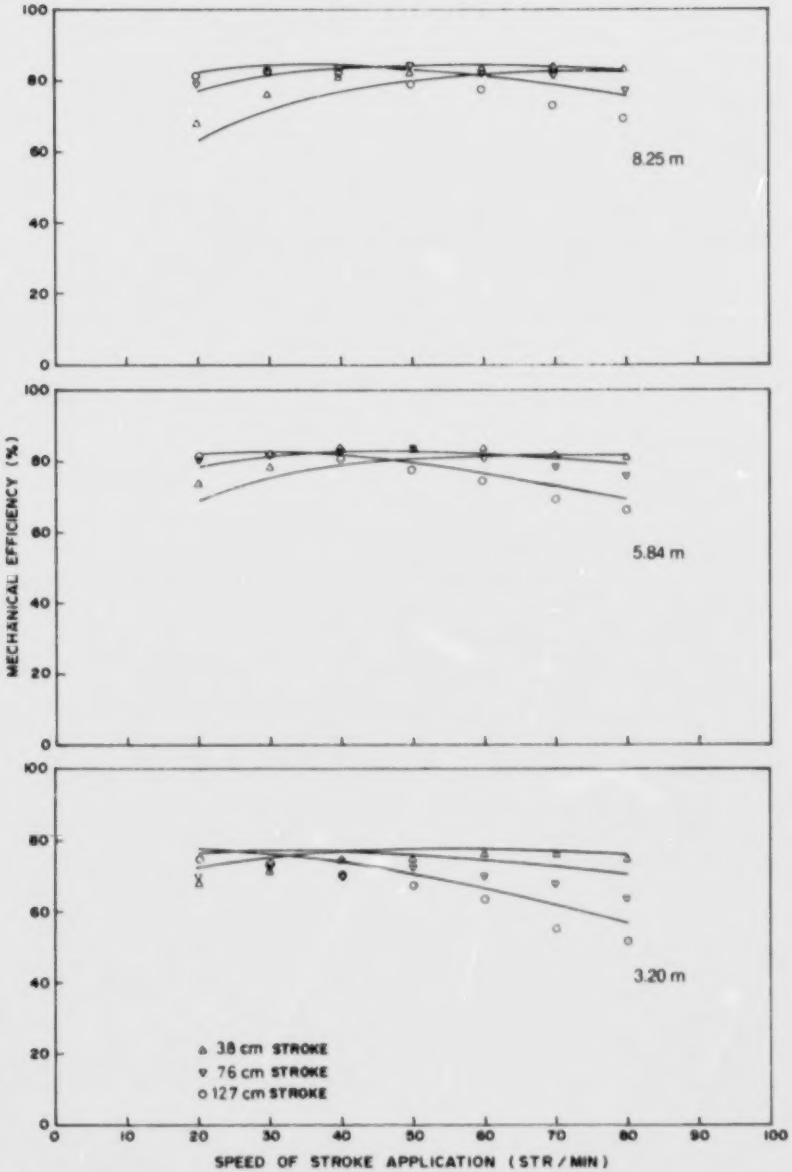

Fig. 27. Variation of mechanical efficiency with speed of stroke application at 8.25, 5.84, and 3.20 m water head (points are experimental data and solid lines are predictions).

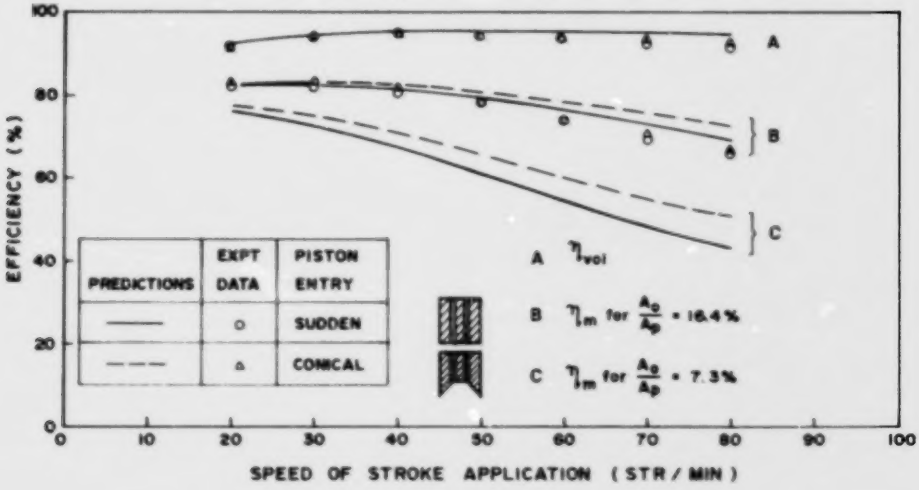

Fig. 28. Effect of conical entry in piston and foot valve on volumetric and mechanical efficiency.

mechanical efficiency improved only 2.5%. This is so in the present case because U_o in equation 12 is small because of the large ratios of orifice to valve area. The pressure loss across the piston and foot valve is, therefore, small and k_c has only a small effect on the mechanical efficiency. On the other hand, if the two area ratios are small, U_o and Δp are both large, and k_c has a more significant effect on the mechanical efficiency. Figure 28 also shows the theoretical predictions of the mechanical efficiency for orifice to valve area ratio of 7.3%. At the highest piston speed, the mechanical efficiency is improved by 7.4% if a conical entry is used in both piston and foot valve.

Effect of size of piston

Experimental results (Fig. 29) show that volumetric and mechanical efficiencies are not significantly affected by a difference between the cylinder diameter (54.48 mm) and the piston size (53.98, 52.71, 51.69, and 50.42 mm) of up to 4.06 mm. This is not unexpected as the water seal between the piston and the cylinder is effected by the piston rings expanding to fit the bore of the cylinder.

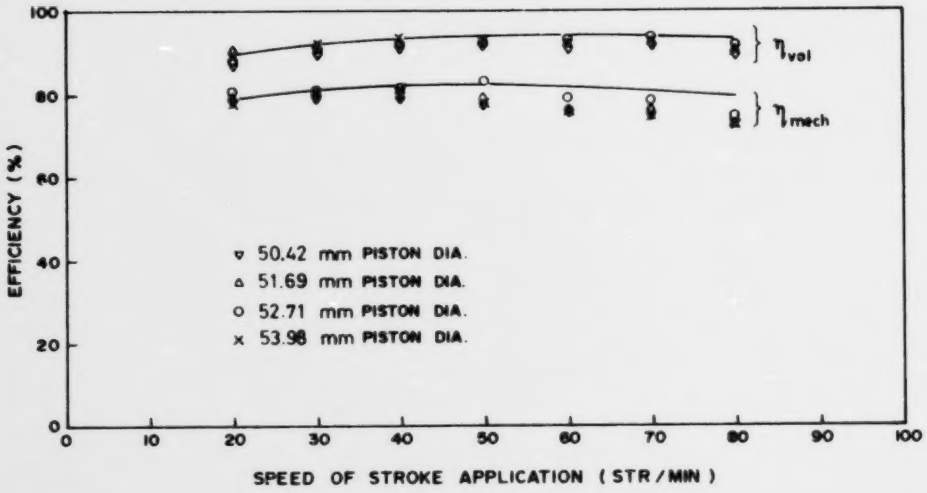

Fig. 29. Variation of volumetric and mechanical efficiencies with speed of stroke application for four piston diameters (points are experimental data and solid lines are predictions).

Effect of thickness and number of rings

Experimental results show that increasing the number of piston rings from two to four has no significant effect on the volumetric and mechanical efficiencies (Fig. 30). The figure also shows variation of volumetric and mechanical efficiencies with speed of stroke application for piston ring thicknesses of 3.18, 6.35, and 12.7 mm. The mechanical efficiencies are independent of the piston ring thickness for the 6.35-mm and 12.7-mm piston rings but 3.18-mm rings resulted in a drop in the volumetric efficiency but an increase in mechanical efficiency as a result of decreased friction between the rings and the pump cylinder.

Computer Simulation of Handpump Performance

The preceding sections show that the predictions using the mathematical model described in chapter 2 agree well with the experimental results. Thus the

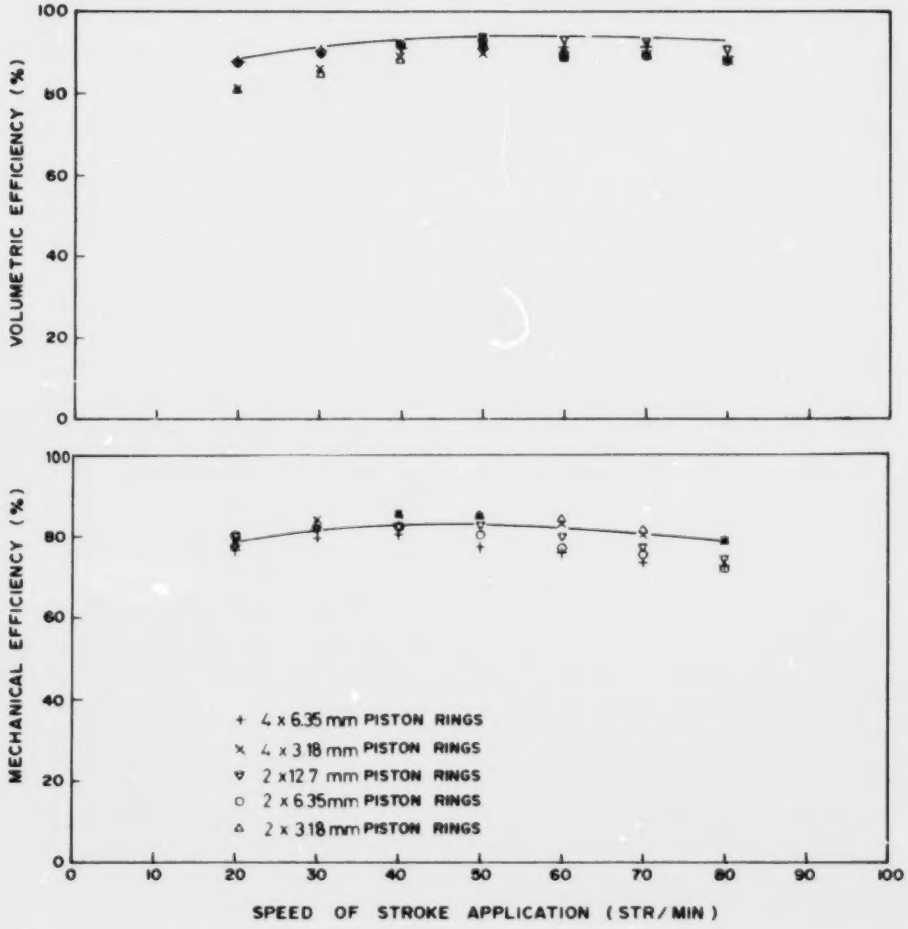
VOLUMETRIC EFFICIENCY (%)
MECHANICAL EFFICIENCY (%)
+ 4 x 6.35 mm PISTON RINGS
× 4 x 3.18 mm PISTON RINGS
▽ 2 x 12.7 mm PISTON RINGS
○ 2 x 6.35mm PISTON RINGS
△ 2 x 3.18 mm PISTON RINGS
SPEED OF STROKE APPLICATION (STR/MIN)

Fig. 30. Effect of thickness and number of rings on the volumetric and mechanical efficiencies (points are experimental data and solid lines are predictions).

effect of the many parameters that affect handpump performance may be further investigated more efficiently by computer simulation without resorting to the more laborious process of running different experiments for the various combinations of experimental parameters.

Optimum Design

Leakage past the piston ring seals, friction between the piston ring seals and the pump cylinder, valve closure delays, and pressure drops across the piston and foot valve have been shown to have a pronounced effect on pump performance. The manner in which these dynamic effects increase or decrease the volumetric and mechanical efficiencies are expressed mathematically in equations 6 and 7. Any detailed design of the piston and foot valve assemblies must consider these dynamic effects to enable the critical dimensions of the design to be optimized.

The ratio of orifice to valve area should be sufficiently large to prevent large pressure drops at the desired speed of operation. For the present valve configuration, a value in excess of 15% is satisfactory. The structural strength of the piston and foot valve design should be considered if large area ratios are being contemplated.

The piston speed, which is a product of stroke length and speed of stroke application, should be sufficiently high to ensure low leakage past the piston rings. A boy operating a handpump is observed to be able to achieve a piston speed of the order of 9 m/min (0.15-m stroke at 30 strokes/min or 0.09-m stroke at 50 strokes/min).

The valve weight must be sufficient to ensure minimal closure delay at the particular valve gap. However, the valve gap must not be too small as it is likely to increase the pressure drop across the piston and foot valve resulting in decreased mechanical efficiency. For a 50-mm diameter pump, a valve flap weight between 13 and 30 g operating with a valve gap of 6 mm gives satisfactory performance.

Increased water head increases the leakage past the piston rings resulting in decreased volumetric efficiency. However, if the pump is operated at high piston speeds, the effect is much reduced.

As the sealing action at the piston is effected by the piston rings expanding to fit the pump cylinder, small dimensional variations of clearance between the

piston and draw pipe have no effect on the performance characteristics. For the 50-mm diameter pump, performance did not deteriorate significantly for a difference in diameter of piston and pipe up to 4 mm.

As the use of a conical entry in the piston and foot valve has only a small effect on the mechanical efficiency at normal operating speeds, the simplicity of the sudden contraction entry may be retained to save cost of manufacture. However, the holes should preferably be slightly chamfered at the entry.

IDRC-UM Handpump Design

Suction or Lift Handpump

In the design of pumps to draw water from a given depth, there is a certain "maximum suction depth" below which it is no longer possible to draw water by suction. Below this depth, water must be raised either by lifting or some other method. This distinction is important as a suction handpump is generally simpler and less expensive than a lift pump.

The major advantages and disadvantages of the present suction and lift handpump designs (Fig. 31) are shown in Table 1.

Main Features of IDRC-UM Handpump

The main features of the IDRC-UM suction and lift handpumps are

. A PVC cylinder fitted with a sliding PVC piston and a stationary, but recoverable, PVC foot valve (Fig. 32). The piston is equipped with two polyethylene ring seals.

. A leverage system consisting of timber linkages and galvanized iron with oil-impregnated timber bearings (Fig. 33).

The main advantages of the plastic handpump over the traditional cast metal design are

. Simple fabrication procedures because PVC

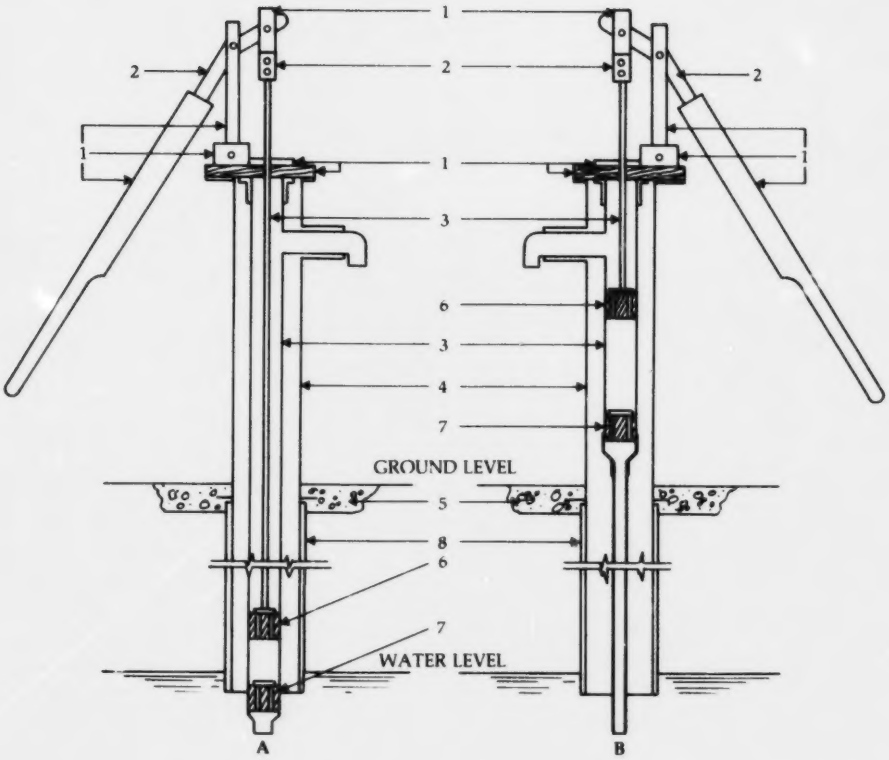

Fig. 31. (A) Lift and **(B)** suction pumps: (1) wooden parts, (2) galvanized parts, (3) PVC parts, (4) mild-steel stand, (5) concrete, (6) piston, (7) foot valve, and (8) casing pipe. (From reference 23, page 44.)

parts can be solvent-welded together. Also PVC parts can be worked on using light hand tools.

. Maintenance by users at the village level becomes feasible because PVC is light, and removal of the hand-pump assembly from the well for inspection and repairs is relatively easy.

. Plastic parts are free from corrosion and rust.

The leverage system is designed to eliminate the requirement for metal journal or roller bearings, which are not readily available in rural areas and which are

Table 1. Comparison of suction and lift handpumps.

Lift handpump	Suction handpump
Main features	
Foot valve must be less than the "maximum suction depth" above the water level and usually below the water level. Piston is about 0.3 m above the foot valve.	Both piston and foot valve are above ground level. Riser pipe below the foot valve can be smaller than pumping cylinder.
Advantages	
Can be used to draw water from depths greater than the "maximum suction depth." Handpump is self-priming.	Easier to install the suction handpump. Handpump cylinder can be assembled at workshop. Piston and foot valve are readily accessible for servicing and repairs.
Disadvantages	
Generally more difficult to install than suction handpump. Foot valve is more difficult to remove for servicing or repairs than in suction handpump.	Cannot be used to draw water from below "maximum suction depth." Handpump requires priming if the foot valve leaks.

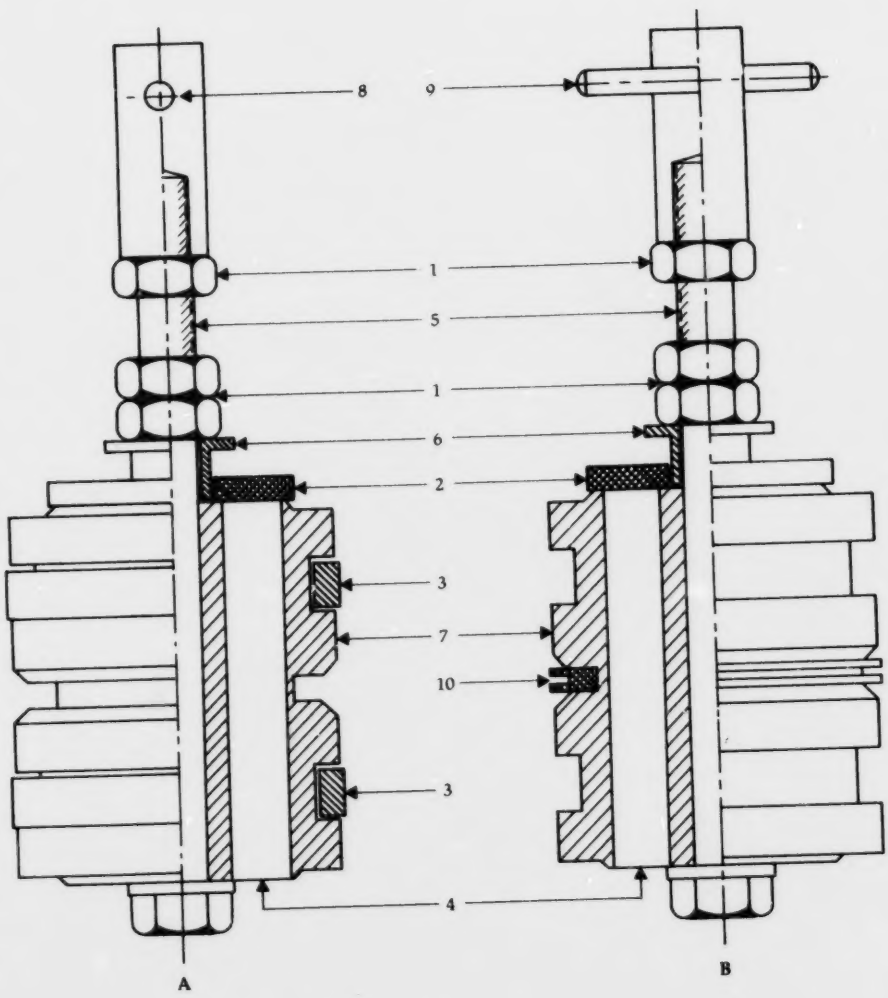

Fig. 32. (A) Piston assembly and **(B)** recoverable foot valve: (1) lock nuts, (2) valve flap (natural rubber), (3) piston rings (polyethylene), (4) six equally spaced holes, (5) bolt, (6) valve stop, (7) PVC plastic, (8) hole for connecting pin, (9) recovery pin, and (10) double-lip rubber seal. (From reference 23, page 45.)

prone to rust when exposed to the weather. Also, replacement of timber parts can be carried out by the villagers themselves using locally available timber.

Fig. 33. The IDRC-UM handpump.

Piston Ring Seal Design

An important innovation of the present design is the use of hard plastic split-ring seals for the piston. These are made with two free ends and with an unsprung diameter slightly larger than the pump cylinder but with a circumferential length slightly shorter than the circumference of the pump cylinder. This allows

the ring seals to be compressed and sprung into the pump cylinder.

The main advantages in the use of these hard plastic split-ring seals over conventional soft seals, such as those of rubber or leather, in the piston are

. The pump can be operated at a higher piston speed and with a smaller frictional resistance.

. As the pistons are made slightly smaller than the cylinder and the fluid seal is achieved by the rings, which expand or contract to fit the variations in the cylinder diameter, the piston and pump cylinder do not have to be made with close dimensional tolerances. This is important because standard grade commercial PVC piping may be used as pump cylinders.

. A certain amount of cylinder wear may be taken up by the ring expanding outwards without any loss in sealing efficiency.

. The latest development in the ring design showed that wear in the PVC cylinder can be reduced significantly.

Recoverable Foot Valve

It is often necessary during maintenance of the handpump to recover the foot valve, which may be a long way down the riser pipe, for inspection and repair and later to replace it. Therefore, it is a major advantage if the foot valve can be removed without having to raise the complete riser pipe. Also, because the pump cylinder is of standard commercial grade PVC pipe, it may have significant variations in dimensions so that the foot valve must not be a tight fit.

The double-lip rubber seal in the foot valve (Fig. 32) allows for these requirements.

First, it is sufficiently flexible to enable the foot valve to be readily pushed into or pulled out from the pump cylinder through the top of the riser pipe using a special extractor hook (Fig. 34).

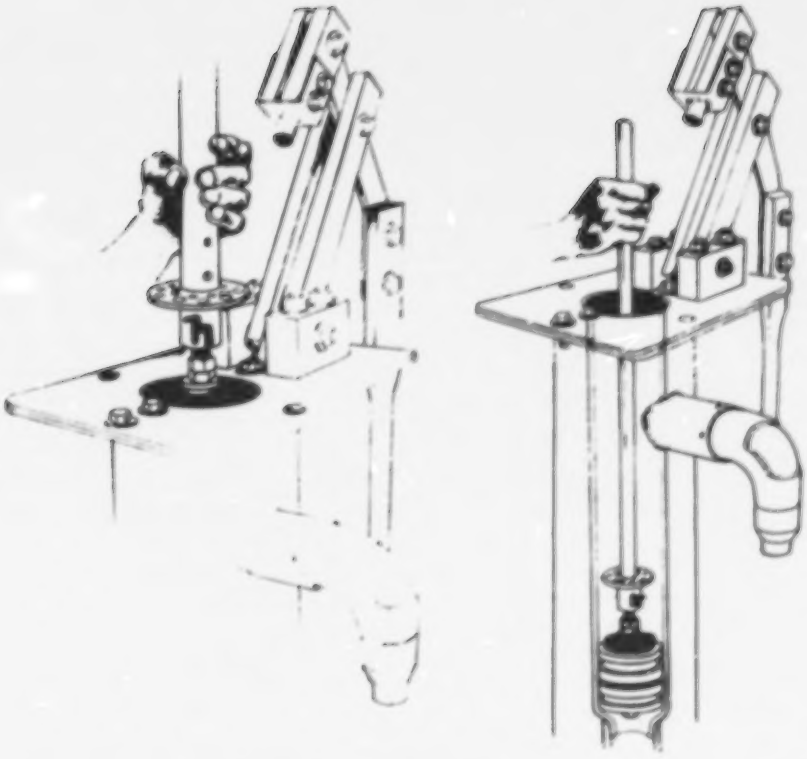

Fig. 34. Extractor rod for removal of foot valve from pump cylinder. (From reference 34, page 12.)

Second, it acts as an effective water-tight seal between the barrel of the foot valve and the pump cylinder.

Third, it keeps the foot valve in a fixed position in the pump cylinder during the pumping strokes.

To facilitate the raising of the foot valve, the pump cylinder (usually 1 m long), which is joined to the riser pipe, is made of a slightly thicker grade of PVC piping than the riser pipe so that there is a clearance between the foot valve double-lip rubber seal and the riser pipe. When the foot valve is pulled clear of the pump cylinder, any water remaining in the pump flows past the foot valve, which greatly facilitates the recovery of the foot valve.

4. FIELD INVESTIGATION

After a handpump has been tested in the laboratory, it must be tested under actual conditions in the field. If there are several national field-testing programs, field methodology and measurement techniques must be standardized to ensure that data collected in one program are compatible with those from another. Ideally, the same data collection sheets should be used for all the programs. If this is not possible, then a minimal set of data to be collected, which is common to all the programs, should be agreed upon.

Data Collection Sheets

Four different sets of forms are used for monitoring the mechanical performance of the handpumps (appendix C). Form A -- Well and pump specification sheet is used at installation of the pump; Form B -- Site visit data sheet is used for monthly visits, and covers mechanical performance, maintenance operations, wear measurements, and user feedback; Form C -- Failures and repairs sheet is used whenever necessary; and Form D -- Handpump caretaker's data collection sheet is filled in weekly.

Methods of Measurement and Measuring Equipment

The method and instrument used often define the accuracy of a measurement. Table 2 lists the recommended equipment and methods of measurement to be adopted for obtaining the data specified in the data collection sheets (appendix C).

Other Measurement Techniques Used in Field

Several of the laboratory measurement techniques used to determine the technical performance of the

Table 2. Field measurement techniques.

Measurement	Method or equipment
Actual volume discharged	Use bucket (minimum: 25 L) or flowmeter
Water head	Use electrode probe and wire attached to an electric multitest meter, sound probe, or similar equipment
Total work output	Use counter at fulcrum or rod
Work input	Use mechanical plotter
Wear measurements	
Journals	Use internal vernier calipers to measure wear at 3 positions at 120° apart at maximum wear cross-section
Shafts and rods	Same as in journals above, but use external vernier calipers
Guides	Same as in journals
Height of valve seat ridge	Place engineer's square on top of valve seat ridge and measure distance between engineer's square and bottom of ridge using feeler gauges or micrometer depth gauge
Piston ring	Measure radial and lateral thickness at 3 points 120° apart at top, middle, and bottom positions using vernier calipers or micrometers
Pump cylinder diameter	At end of field test, cut cylinder into 2 halves along its longitudinal axis and measure thickness using a

(continued)

Table 2. Concluded.

Measurement	Method or equipment
	micrometer mounted on a special "U" bracket
Piston	Initial dimensions of (a) width and depth of grooves and (b) diameters at appropriate positions to be measured before installation using micrometers or vernier calipers
	Subsequent readings taken during site visits

reciprocating piston pump cannot be used in the field either because sophisticated electronic equipment is needed or because of physical constraints at the hand-pump site. Therefore, several simple field measurement techniques and instruments were developed for use in the field monitoring program.

Depth of Water Table

The technique used to measure the water level in the well relies on the difference in the electrical resistance of water and air. The equipment consists of a conical perspex probe head (Fig. 35) on the end of a sufficient length of twin-core wire to reach the water level from the top of the pump. The twin wires are threaded into the central hole of the probe head and the ends of the wires pass through two lateral holes to emerge flush with the surface on either side of the probe head. The probe head is machined with a total conical angle of 20° and the surface polished to facilitate drip-dry action when the probe is lifted out of the water. A multitest meter capable of measuring up to 300 kΩ is connected to the other end of the twin-core wire to measure the resistance across the two terminals. When the terminals are exposed to air, the meter reads open circuit but when they are immersed in water, the meter reads in the order of 200 kΩ. Discrete changes from infinity to 200 kΩ can be obtained

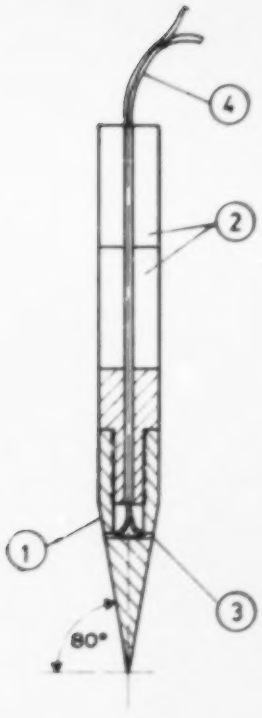

Fig. 35. Probe for measuring well depth: (1) probe head, (2) brass weights, (3) two lateral holes for twin-core wire, and (4) twin-core wire. Probe diameter is 12.7 mm.

on the meter for changes in the level of water of less than 6 mm. The depth of the water table from the ground can be determined from the length of the wire.

Total Usage of Handpump

For a comparative study of wear and physical deterioration of major handpump components, the total usage of the handpump in the field over the period must be measured.

We adapted a Veeder-Root totalizing counter (which does not register reversed rotation) to measure the cumulative travel of the piston rod during the useful

stroke (Fig. 36). The mounting is made from a 1.27-cm thick plate. After drilling the central hole (slightly larger than the piston rod), the plate is cut in half. A third similar piece is required for mounting the assembly so that the movable half-piece is clear of the top-plate of the handpump. The Veeder-Root counter is mounted on the movable half-piate, which is spring-loaded to ensure constant contact of the roller with the piston rod. The fixed half-plate should be mounted clear of the piston rod allowing for any swing or inclination of the piston rod during the applied stroke.

Work Input

The method used in the laboratory to determine work input is not suitable for use in the field because of the difficulty of transporting several pieces of very sensitive electronic equipment and ensuring that they remain in good operating condition at the field site. Moreover, the method also requires a highly trained technician to carry out the measurements. It is, therefore, desirable to have a simpler, lighter and more robust instrument that can be readily carried to the field and operated by a technician with a lower level of training (24).

Therefore, a mechanical instrument was used to measure the work input at the field site (Fig. 37). It consists of

- Two steel bars held together at both ends by adjustable clamps -- the bars deflect relative to each other when a load is applied to the midpoints of the bars.

- A dial gauge that magnifies the deflection of the steel bars.

- An indicator drum that rotates the tracing paper as the instrument is displaced from a fixed position.

- A pen connected to the dial gauge by a string that is held taut by a recoiling spring.

During the cycle of operation of the reciprocating water pump, the relative deflection of the bars after

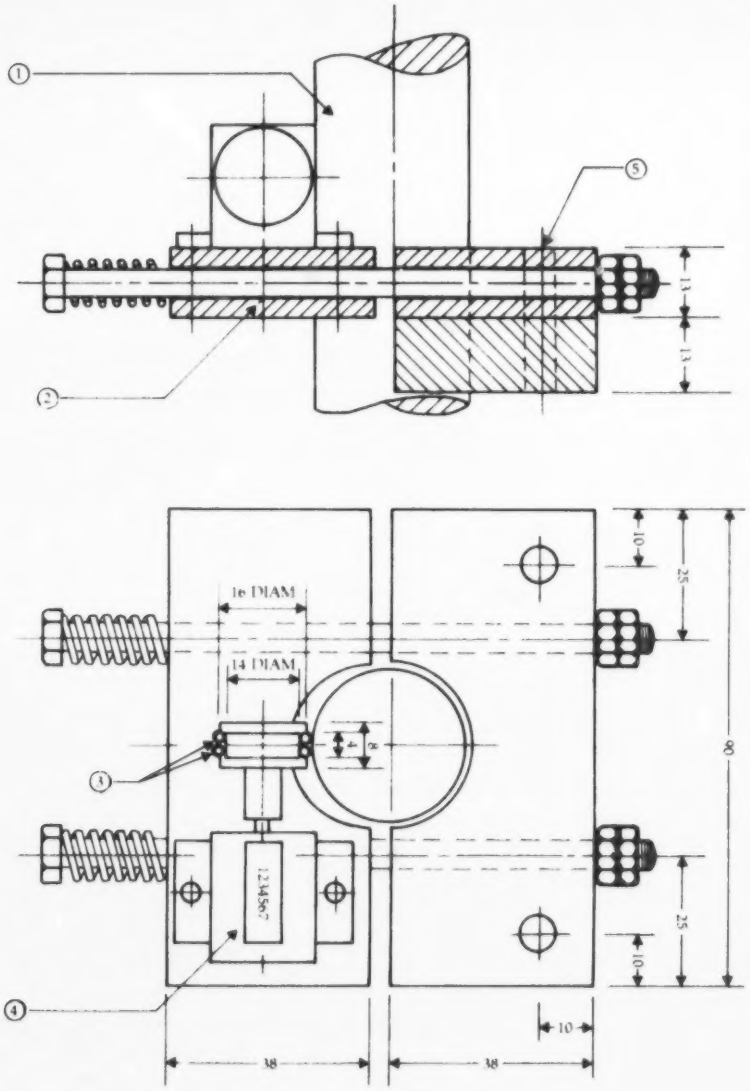

Fig. 36. Mounting for Veeder-Root (V-R) counter on handpumps (all dimensions in mm): (1) piston rod, (2) sliding fit to allow for lateral movement of counter, (3) two O-rings (13 mm ID x 2 mm diameter cross section), (4) V-R flange counter (no. 74-6125-001), (5) mounting screw to top plate of hand pump, and (6) bearing blocks.

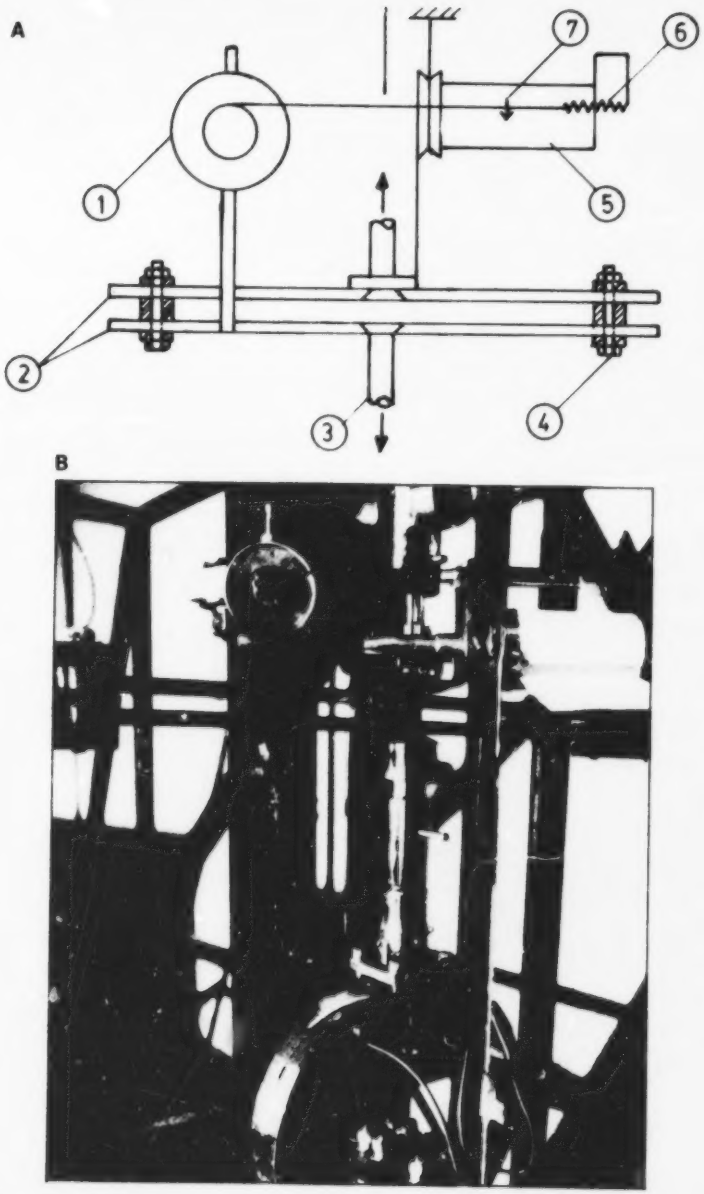

Fig. 37. (A) Schematic of the mechanical plotter: (1) dial gauge, (2) mild steel bars, (3) piston rod, (4) adjustable clamps, (5) indicator drum, (6) spring, and (7) pen. **(B)** A mechanical plotter for measuring work input.

being magnified by the dial guage causes the pen to move perpendicular to the rotation of the drum, which is rotated by displacement of the instrument from a given fixed position. The resultant diagram traced on the paper on the drum gives a force-displacement loop that is equivalent to that obtained by the laboratory method (Fig. 38A).

The operating range of the mechanical instrument with respect to the applied force may be changed by changing the distance between the adjustable clamps so that the effective deflecting length of the steel bars is either reduced or increased.

Comparison of the force-displacement loops obtained by the oscilloscope or "electronic" method and by the use of the mechanical plotter (Fig. 38) show remarkable similarity and that the main characteristics of the loop are adequately reproduced by the mechanical plotter.

The maximum difference in work input, computed from the area enclosed by the force-displacement loop (for a range of operating speeds at three water heads) measured by the mechanical plotter and by the electronic method, was 4% (Table 3), which is within the experimental error that is to be expected in a field measurement.

Results of Field Monitoring Program

Seventeen handpumps were installed and used by villagers for about 4 years. However, the handpumps were fully monitored only for an 8.5-month period. Samples of performance results computed from field data collected for the 17 handpumps are presented in the following sections.

Tabulated Performance and Maintenance Data

To permit comparison among handpumps, it is useful to provide a summary of data for all handpumps in a single table (Table 4) although, inevitably, only a single average value of each performance criterion can be included in such a table. A summary of maintenance

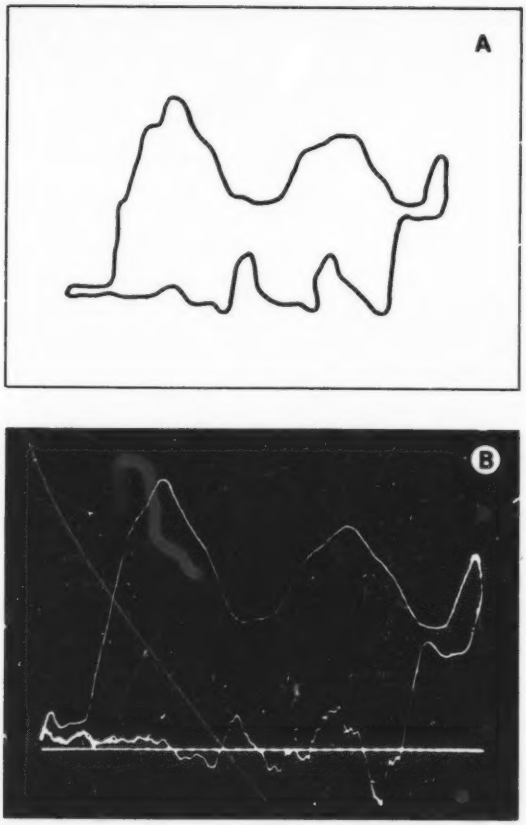

Fig. 38. Force-displacement diagram for 5.8 m water head and 60 strokes/min with a large valve delay recorded by **(A)** mechanical and **(B)** electronic methods.

and repairs carried out on all 17 handpumps is presented in Table 5.

Variation with Calendar Time

Volumetric efficiency

From equation 27, we have

$$\eta_{vol} = 1 - x/L_o - C_1 d_p h_T/(NL_o A_p)$$

$$= 1 - x/L_o - C_1 C_2 h_T$$

Table 3. Comparison of work input measured by mechanical and electronic instruments.

Speed (strokes/min)	Work input (J)		Difference (%)
	Mechanical	Electronic	
	3.2-m water head		
20	15.2	15.7	3.2
30	15.7	16.0	1.9
40	16.4	16.3	0.6
50	17.1	17.1	0.0
60	17.5	17.2	1.7
	5.8-m water head		
20	27.7	27.3	1.4
30	27.7	27.3	1.4
40	27.9	27.3	2.2
50	28.5	27.3	4.2
60	28.5	27.3	4.2
	8.2-m water head		
20	35.5	35.4	0.3
30	35.5	35.4	0.3
40	35.4	35.8	1.1
50	35.5	36.3	2.3
60	36.2	37.7	4.1

where

$C_2 = d_p/(NL_oA_p)$

For constant N and L_o and small values of valve delay (x/L_o), we get

$\eta_{vol} = 1 - C_1C_2h_T$

This equation shows that the volumetric efficiency is a function not only of the water sealing term (C_1C_2) but also of the water head (h_T).

Table 4. Summary of performance data.[a]

Pump number	Type[b]	Date installed	Number of users (persons)	Efficiency (%) η_{vol}	Efficiency (%) η_{mech}	Water head (m)	Average water output (L/day)	Down time (days)	Maintenance and repairs: Parts (US$)	Maintenance and repairs: Time (hours)	Period monitored
PK 001S	S	19/11/80	50	95.0	-	2.6	381.0	0	9.40	0.50	6/7/81-16/3/82
PK 002S	S	18/11/80	40	98.0	-	2.4	58.9	0	0.26	0.50	6/7/81-16/3/82
PK 003S	S	19/11/80	40	95.0	-	2.8	93.8	0	-	-	7/7/81-16/3/82
PK 004L	L	2/12/80	30	89.0	-	2.5	502.7	0	-	-	7/7/81-17/3/82
PK 005L	L	2/12/80	50	98.0	-	2.7	186.2	0	8.45	1.50	7/7/81-17/3/82
PK 006S2	S	3/2/81	40	84.0	-	3.5	217.6	60	-	2.50	7/7/81-17/3/82
NS 001S	S	6/7/80	25[c]	83.5	59.6	3.7	487.5	3	-	0.50	25/6/81- 7/4/82
NS 002S	S	6/7/80	25[c]	93.7	59.1	3.7	133.5	0	-	-	25/6/81-12/4/82
NS 003L	L	24/7/80	25[c]	92.0	-	9.2	152.2	4	0.50	3.00	29/7/81-19/4/82
NS 004S	S	24/7/80	25[c]	79.2	64.0	7.0	85.2	0	-	-	29/7/81-14/4/82
NS 005S	S	28/7/80	25[c]	84.4	62.2	6.9	59.9	0	-	-	11/8/81-19/4/82
NS 006S	S	28/7/80	31	81.7	65.0	4.9	351.3	0	9.40	0.17	17/6/81-12/4/82
NS 007PS	PS	18/9/80	40	75.0	-	4.9	524.0	1	10.50	2.75	17/6/81- 9/4/82
NS 008PL	PL	25/9/80	50[c]	90.0	-	5.4	826.8	0	7.80	3.00	22/7/81- 9/4/82
NS 009L	L	2/10/80	15[c]	70.5	65.2	7.6	242.6	10	0.75	21.75	21/10/81-16/4/82
NS 010S	S	9/10/80	50[c]	95.0	70.4	4.6	718.2	0	-	-	15/6/81- 7/4/82
NS 011PS	PS	16/10/80	35[c]	79.0	-	5.8	506.0	4	12.00	7.50	20/7/81-14/4/82

[a]Source: Reference 23, page 48.

[b]S, suction; L, lift; PS, pressure-suction; PL, pressure-lift

[c]The number of households rather than number of users was given. An average of five persons to a household is used as an estimate.

Table 5. Summary of maintenance and repairs for pumps at Perak (PK) and Negri Sembilan (NS).

Pump number	Date installed	Maintenance date	Down time (days)	Description	Cost of spare parts (US$)	Time required (hours)
PK 001S	19/11/80	27/1/81	0	Broken PVC spout replaced	9.40	0.50
PK 002S	18/11/80	27/1/81	0	Leaking foot valve changed to new Linard rubber valve flap	0.26	0.50
PK 003S	19/11/80		0	None	-	-
PK 004L	2/12/80	27/1/81	0	Wood cover cracked but no action taken	-	-
PK 005L	2/12/80	10/9/81	0	Fulcrum too loose and retightened	-	-
				Piston bolt broken during dismantling and replaced	8.45	0.50
				PVC piston rod cracked at brass pin hole; relocate new hole	-	1.00
PK 006S2	3/2/81	7/7/81	60	Many problems due to well having insufficient water; lift pump replaced with suction pump	-	2.00
		10/9/81	0	Fulcrum arm/base too tight and reset	-	0.50
		10/9/81	0	Wood cover piece cracked but not replaced	-	-
NS 001S	6/7/80	30/9/81	0	Piston and foot valve cleaned of iron stains	-	0.50
		3/2/81	3	Priming inlet not closed properly	-	-
NS 002S	6/7/80		0	None	-	-

NS 003L	24/7/80	28/9/81	4	Piston rod broke at socket joint	0.50	3.00
NS 004S	24/7/80	–	0	None	–	–
NS 005S	28/7/80	–	0	None	–	–
NS 006S	28/7/80	25/11/81	0	Spout missing	9.40	0.17
NS 007PS	18/9/80	20/2/81	1	Broken brass piston-connecting bolt replaced with mild steel (chromed) part	6.00	2.00
		23/4/81	0	Foot-valve brass bolt broken	4.50	0.50
		17/6/81	0	Wooden fulcrum arm/base loose and retightened	–	0.25
NS 008PL	25/9/80	20/2/81	0	Replaced brass connecting bolt with mild steel part (no sign of damage)	6.00	1.00
		19/3/81	0	Brass hinge joint badly worn; hinge eliminated from the design	–	1.75
		30/9/81	0	Wood fulcrum arm cracked	1.80	0.25
NS 009L	2/10/80	19/12/80	0	Leaking foot valve replaced with Linard rubber valve flap	0.25	6.75
		9/6/81	10	PVC piston rod broken	0.50	15.00
		19/8/81	0	Nuts of wood fulcrum loose due to wear of the wood and retightened	–	–
NS 010S	9/10/80		0	None	–	–
NS 011PS	16/10/80	21/1/81	1	Broken brass connecting bolt replaced	6.00	2.00
		13/2/81	3	Same part broken again and replaced	6.00	3.00
		20/2/81	0	Replaced above part with mild steel part	–	2.50

Source: Reference 23, pages 48–49

Figure 39 is an example of the variation of efficiency (line 1) and water head (line 2) with calendar time. To isolate the effect of the water head on the volumetric efficiency, plots of $(1 - \eta_{vol})/h_T$ against calendar time were made for each of the handpumps (for example, Fig. 40). There is considerable experimental scatter for small values of $(1 - \eta_{vol})/h_T$, which is equivalent to high values of η_{vol} and small values of h_T. For larger values of $(1 - \eta_{vol})/h_T$, the experimental points show a more consistent trend (Fig. 40, closed triangles). (The straight lines through the experimental points were plotted using the method of least squares.) For most pumps, the expression $(1 - \eta_{vol})/h_T$ decreases with time indicating that the sealing efficiency improves with time.

Water head

The water head curves (Fig. 39, line 2) in each district show a similar pattern of variation with calendar time indicating the dry and wet seasons in the respective areas.

Average daily water delivery

The average volume of water delivered per day is computed from the counter readings, which is the sum of the piston travel, and the average volumetric efficiency for the period between monitoring visits. The average volume of water delivered per day varies from handpump to handpump as well as for one handpump at different times of the year. It is interesting to note that the average volume of water delivered per day is higher when the water head is greater, which coincides with the dry seasons for the particular district (Fig. 39, lines 3 and 2).

Total Work Output

The total, or accumulated, work output is computed as the product of the average volume of water delivered per day and the average water head for the period between monitoring visits. The total work output varies considerably between different handpumps either because some are used more than others or because of a difference in water head. One would expect the general wear and tear of the handpumps to

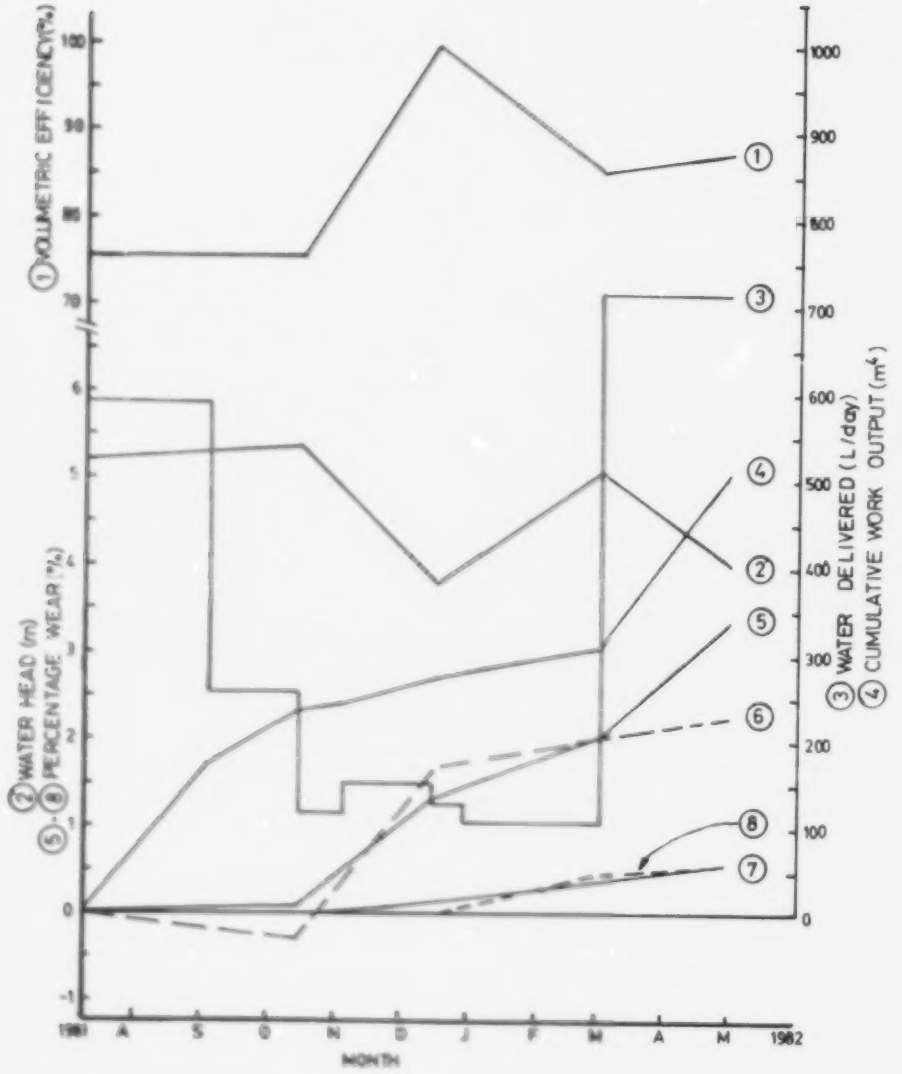

Fig. 39. Performance characteristics of handpump NS 006S in field test: (1) volumetric efficiency, (2) water head, (3) water delivered per day, (4) cumulative (total) work output (m^4 = m^3 [volume] x m [lift]), percentage wear of (5) top wooden bushing (0°), (6) bottom wooden bushing (90°), (7) top piston ring, and (8) bottom piston ring.

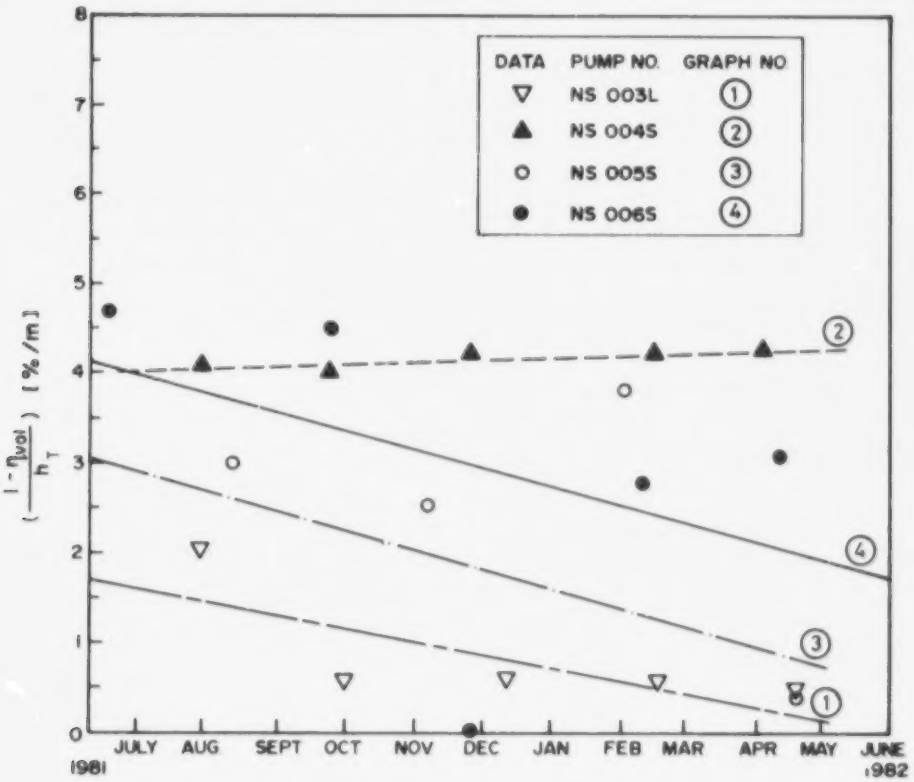

Fig. 40. Variation of $(1 - \eta_{vol})/h_T$ with calendar time for four sample handpumps.

be more dependent on the total work output rather than the calendar time.

Variation of Wear with Calendar Time and Total Work Output

Although wear measurements were made on a number of components, significant wear was found only in the top wooden bush supporting the piston rod, the piston rings, the handpump cylinder, and the brass pins used in pin-joints of the piston rod (however, this was not measured).

Wear in bore of top wooden bush

Wear in the bore of the wooden bush was calculated as a percentage from change in diameter divided by original diameter and the angular position on the hole was defined with reference to the plane of the lever, i.e., $\theta = 0°$ for the diameter in the plane of the lever and $\theta = 90°$ for the diameter perpendicular to it.

Wear in the bore of the wooden bush can be plotted against calendar time, e.g., Fig. 39 (lines 5 and 6), or against total work output, e.g., Fig. 41. As expected, the wear is more dependent on the total work output than on the calendar time. The magnitude of the wear also varies from handpump to handpump depending not only on the total work output but also on the dampness of the wooden bush and the slack in the lever system. It is also not surprising that the wear in the plane of the lever ($\theta = 0°$) is greater than that perpendicular to it ($\theta = 90°$).

Wear of piston rings

Wear of the high density polyethylene (HDPE) piston rings as a percentage was calculated as change in thickness divided by original thickness.

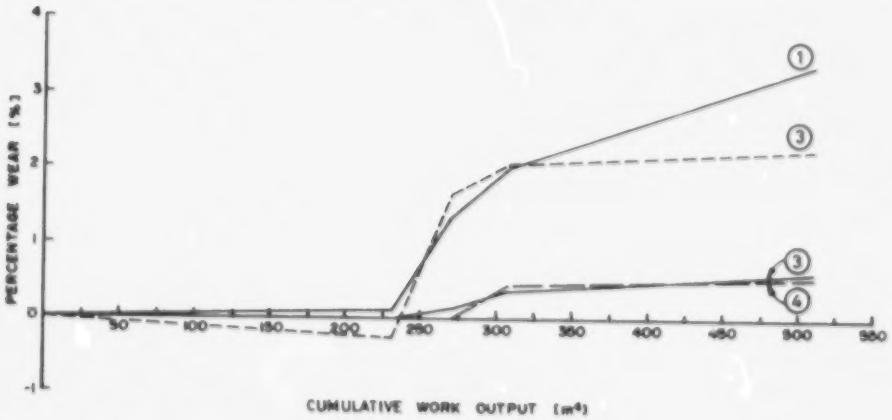

Fig. 41. Variation of wear with cumulative (total) work output for handpump NS 006S: (1) top bushing (0°), (2) bottom wooden bushing (90°), (3) top piston ring, and (4) bottom piston ring.

Again, the wear of the piston rings is more a function of total work output (Fig. 41, lines 3 and 4) than the calendar time (Fig. 39, lines 7 and 8). The average total wear of the piston rings over the 8.5-month period is of the order of 4%.

Wear of the pumping section of the handpump cylinder

The PVC cylinders of five of the handpumps were removed and cut in half along the longitudinal axis so that the wear on the cylinder section could be measured in the laboratory. Wall thickness varied considerably around the pumping section of the PVC cylinder after the pumps had been in the field for approximately 2 years (Fig. 42). Figure 43 is a plot of the wear, presented here as the total volume of material removed by wear, against total accumulated sliding distance.

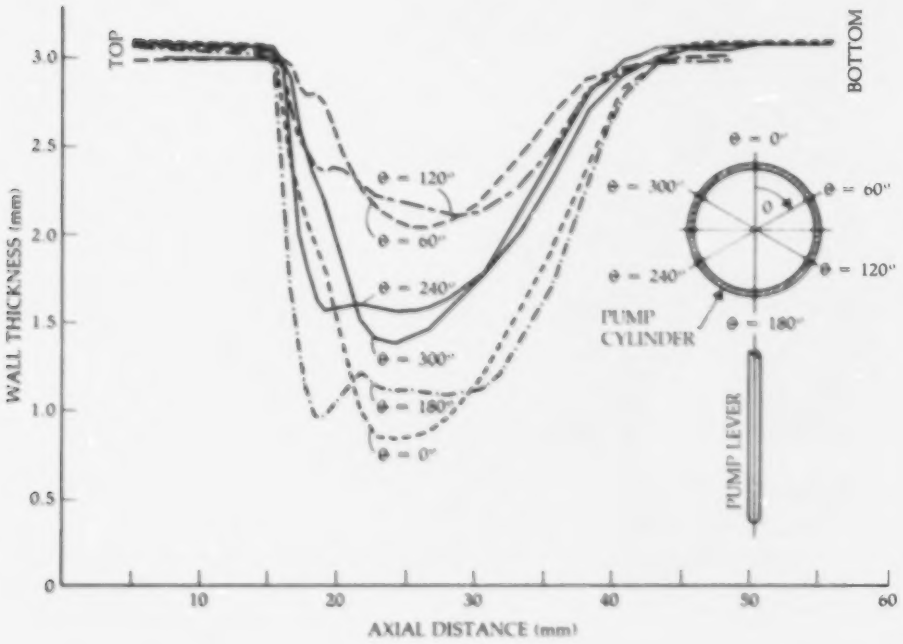

Fig. 42. Variation of wall thickness of pump cylinder with axial distance for pump NS 010S. (From reference 23, page 51.)

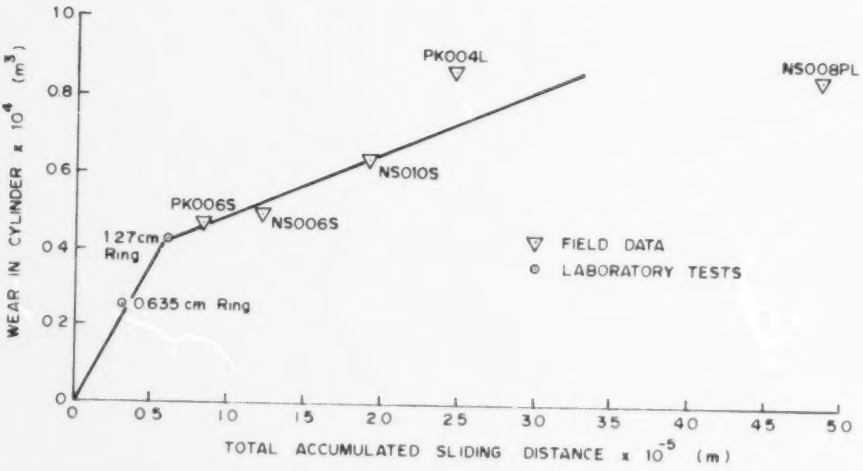

Fig. 43. Variation of wear of PVC cylinder with total accumulated sliding distance.

Because field monitoring was carried out for an 8.5-month period only, the total accumulated sliding distance was obtained by extrapolation using the average accumulated sliding distance computed from counter readings over the monitored period. The rate of cylinder wear was high initially but stabilized to a lower but constant rate. The piston rings are not perfectly circular because they are formed by cutting out a small segment from a circular ring that is of a slightly bigger diameter than the pump cylinder. "Bedding-in" of the piston ring seals may have contributed to the initial higher rate of wear observed in the field data.

The cylinders of handpumps NS 008PL and PK 004L have worn through causing water to leak through the cylinders and the cylinder walls have worn so thin that they have become flexible. This may account for the spread in the wear values for these two handpumps.

Improvements to Wear Characteristics of IDRC-UM Handpump

The excessive wear of the PVC cylinder when fine sand particles are present in the water is a major

problem. In the present design, it was expected that the softer polyethylene ring would wear in preference to the harder PVC cylinder. This was true if the rings were rubbed against the PVC cylinder in clean water. However, when fine sand particles were present in the water, wear on the PVC cylinder was very pronounced whereas no significant wear was observed on the piston rings (30).

A simple solution would be to switch the materials of the rings and the cylinder: the PVC rings rubbing against the polyethylene cylinder would wear off much faster and could be replaced readily. Unfortunately, such an arrangement has disadvantages because polyethylene cannot be solvent-welded and therefore the pump cylinder would have to be designed with special threaded couplings.

The other solution is to improve the design of the ring to reduce the wear.

Comparison of Field Wear Data with Laboratory Test Results

The experimental rig for testing handpumps in the laboratory, which is described in chapter 3, was used to investigate wear of the PVC cylinder when sand particles were present in the water. Initially, 1.27-cm thick HDPE piston ring seals similar to those used in the field tests were fitted to the pistons and measured wear after 400,000 strokes fell on the wear curve of the field results (Fig. 43). A further test using 0.635-cm thick HDPE piston rings, with a stainless steel wire spring inserted in a groove on the inside of the ring to compensate for the inevitable reduction in spring tension, showed a reduction in wear after the same number of strokes. This was to be expected as the effective accumulated sliding distance is half that for the 1.27-cm thick piston rings.

When 1.27- and 0.635-cm thick ring seals were used at two different stroke lengths, the mechanical efficiency of the handpump fitted with 0.635-cm thick piston seals was not significantly different from that of the handpump fitted with 1.27-cm thick piston ring seals (Fig. 44).

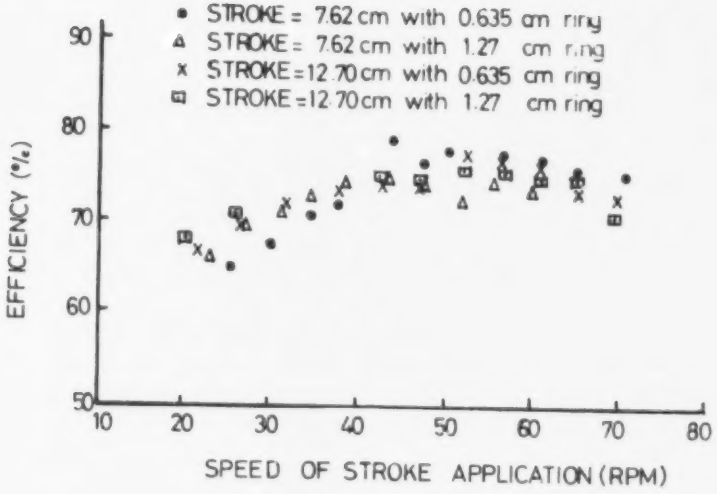

Fig. 44. Variation of mechanical efficiency with speed of stroke application at 6.12-m water head.

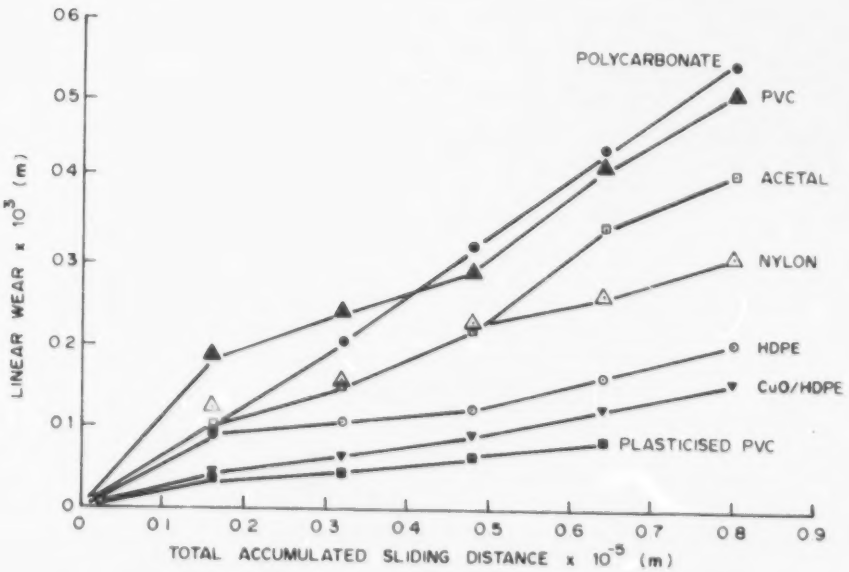

Fig. 45. Variation of wear of PVC cylinder with total accumulated sliding distance for different ring materials.

Variation of Wear of PVC Cylinder with Different Ring Materials

A laboratory investigation was also carried out to determine the rate of wear of the PVC cylinder when ring seals were made from different materials. In this experiment, ring specimens were rubbed against flat PVC pieces in horizontal motion in water containing fine sand particles (27). The ring specimens were made from polycarbonate, PVC, acetal, nylon, HDPE, a combination of HDPE with copper oxide, and plasticized PVC.

With the exception of the last two materials, the others caused more wear than HDPE (Fig. 45). A plausible explanation of the behaviour of polymer-sand-polymer wear is that the sand particles, when trapped between two polymer surfaces in sliding contact, become embedded in the softer of the two polymer surfaces forming an abrasive surface that wears the harder polymer surface (30). When a combination of HDPE and copper oxide is rubbed against PVC, the copper oxide is shed and acts as a solid lubricant that prevents the formation of the abrasive surface. Plasticized PVC, on the other hand, behaves differently. Because of its rubber-like property, it is difficult for the sand particles to get embedded in the plasticized PVC surface, hence no abrasive surface is formed. It also reduces the abrasive action by "giving-way" when pressure is applied and by expelling the sand particles from its surface when the pressure is released.

Recommendations on Improved Ring Design

The results showed that wear in the PVC handpump cylinder may be reduced by decreasing the piston ring thickness and by selecting different ring seal materials, such as a combination of HDPE and copper oxide or a plasticized PVC. Depending on the chosen ring seal material, it may be necessary to modify the ring seal design to maintain the high mechanical efficiency of the handpump.

5. CONCLUSION

It is no longer necessary to design handpumps by trial and error. The mathematical tools are now available in the form of a prediction procedure to determine the volumetric and mechanical efficiencies so that detailed handpump valve designs can be developed through computer simulation.

However, experimental measurements in the laboratory are still required to obtain several physical empirical constants employed in the prediction procedure. Laboratory endurance tests are also needed to determine the endurance life of various parts of the handpump assembly under simulated conditions.

Finally, the prototypes must be subjected to field trials not only to test the technical performance of the handpumps but also to study other socioeconomic aspects such as methodologies to promote community acceptance, community financing and self-help schemes, instructional packages for installation, and maintenance and repairs for village level use.

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APPENDIX A

Interactive Computer Program for Prediction of Handpump Performance

This computer program[1] is written in Applesoft Basic for use in Apple II-compatible microcomputers. The computations are based on the theoretical analysis given in chapter 2.

Specification of Empirical Constants

CLEAK = C_1 in equation 27

The value of CLEAK should be obtained experimentally as described in the second section of chapter 3. This applies not only to the present plastic ring seal design but also to other designs such as the leather or rubber cup seals commonly used in other handpumps.

K1, K2, FKC = k_1, k_2, and k_c in equation 12

ALPHA = α in equation 13

The given values of these constants apply only for valve flaps with a flat design. For conical or spherical plunger valve designs, other appropriate values should be used.

FF = F_T in equation 7

This should be determined experimentally for each design of the piston seals.

---

[1]The program is available from the author on floppy disk.

Specifications of Parameters

DP	=	diameter of cylinder
PSTL	=	length of piston
NH	=	number of holes in piston
DR	=	diameter of holes in piston
VO	=	outer diameter of valve flap
VI	=	inner diameter of valve flap
VGAP	=	valve gap
AMAS	=	mass of valve flap
DEPTH	=	water head
STR	=	stroke length
RPM	=	speed of stroke application

Listing of Computer Program TMODEL

```
10  REM   AN APPLESOFT BASIC INTERACTIVE PROGRAM TO
11  REM   CALCULATE VOLUMETRIC AND MECHANICAL EFFICIENCIES
12  REM   IN RECIPROCATING WATER PUMPS (IN SI UNITS)
15  REM
16  REM
20  REM  * SPECIFICATION OF PHYSICAL CONSTANTS
21  REM
22 RHO = 1000.
23 VSC = 1.116E - 3
24 SG = 1.18
25 PI = 3.142
26 GV = 9.81
27  REM
30  REM  * SPECIFICATION OF EMPIRICAL CONSTANTS
31  REM
32 CLEAK = 3.08E - 3
33 K1 = 0.63
34 K2 = 0.60
35 FKC = 0.45
36 ALPHA = 2
37 FF = 15.6
38  REM
40  REM  * SPECIFICATION OF PARAMETERS
41  REM
42 DP = 0.054
43 PSTL = 0.0762
44 NH = 6
45 DR = 0.009
47 VO = 0.04
48 VI = 0.011
49 VGAP = 0.00287
```

```
50 AMAS = 0.0135
51 DEPTH = 5.84
52 STR = 0.0762
53 RPM = 50
100  REM
151  REM
152  REM    * SPECIFICATION OF CONTROLS
153  REM
154 DT = 0.005
155  REM
156  REM    * SUB 1000-1370 ENABLES PARAMETERS TO BE
     CHANGED
157  GOSUB 1000
158  REM
159  REM    * PRELIMINARY CALCULATIONS
160  REM
161 AV = PI * 0.25 * (VO * VO - VI * VI)
162 AP = PI * 0.25 * DP * DP
163 C1 = .65 * ALPHA * AV * RHO / AMAS
164 R1 = VI / 2
165 R2 = VO / 2
166 AO = NH * PI * 0.25 * DR * DR
167 AN = PI * 0.25 * (DP * DP - VO * VO)
168 AH = PI * VO * VGAP
169 QP = AO / AP
170 QH = AO / AH
171 QN = AO / AN
177  REM
300  REM
301  REM  * COMPUTATION OF VALVE DELAY
302  REM
303 RAD = STR / 2
310 THALF = 30 / RPM
320 TLEAK = CLEAK * DP * DEPTH / RPM
330 OM = PI * RPM / 30
340 IU% = 0
350 T = 0.0
360 DF = 0.
370 T = T + DT
380  IF T > 0.3 THEN  GOTO 750
390 PDF = DF
400 OT = OM * T
401 TIN =  SIN (OT) * RAD / STR
402 TTIN = STR * (1. -  SQR (1. - TIN * TIN))
410 Z = RAD * (1. -  COS (OT)) + TTIN
411  IF IU% = 1 THEN Z = RAD * (1. -  COS (OT)) - TTIN
420 C3 = GV * (1. - 1. / SG)
```

```
430 TXP = T *  SQR (C3 * C1)
435 XAA = .5 * ( EXP (TXP) +  EXP ( - TXP))
440 X =  LOG (XAA) / C1
450 DF = VGAP - X - Z
460  IF IU% = 1 THEN DF = DF - TLEAK * T / (THALF * AP)
470  IF DF > 0. THEN  GOTO 370
480 T = T - DT + DT * PDF / (PDF - DF)
490 OT = OM * T
491 TIN =  SIN (OT) * RAD / STR
492 TTIN = STR * (1. -  SQR (1. - TIN * TIN))
500  IF IU% = 1 THEN  GOTO 540
510 ZU = RAD * (1. -  COS (OT)) - TTIN
520 IU% = IU% + 1
530  GOTO 350
540 ZD = RAD * (1. -  COS (OT)) + TTIN
550 XL = (ZU + ZD) / STR
600  REM
601  REM
602  REM  * CALCULATION OF PRESSURE COEFF FPC
603  REM
610 REY = PI * RHO * STR * RPM * DR / (VSC * 60)
620 FTUR = 0.316 / (REY ^ 0.25)
630  IF REY < 3000 THEN FTUR = 64 / REY
640 FPC = FKC + (1 - QP) * (1 - QP) + FTUR * PSTL / DR
641 FPC = FPC + QN * QN / (K1 * K1) + QH * QH / (K2 * K2)
700  REM
701  REM  * CALCULATION OF EFFICIENCIES
702  REM
710 EV = 1 - XL - TLEAK / (STR * AP)
715 S1 = STR * OM * AP / AO
720 WPRES = FPC * RHO * STR * S1 * S1 / 6
725 WLFT = RHO * DEPTH * STR * GV
730 WNPUT = WLFT * AP * (1. - XL) + FF * STR + WPRES * AP
735 WOUPUT = WLFT * AP * EV
740 EM = WOUPUT / WNPUT
741 EV =  INT (EV * 10000 + 0.5) / 100
742 EM =  INT (EM * 10000 + 0.5) / 100
750  REM
800  REM
801  REM  * PRINT OUT RESULTS
802  REM
810  PRINT ""
815  PRINT : PRINT "THE TOTAL VALVE DELAY IS "XL""
821  PRINT ""
825  PRINT : PRINT "THE VOLUMETRIC EFFICIENCY IS "EV" %"
826  PRINT ""
830  PRINT : PRINT "AND MECHANICAL EFFICIENCY IS "EM" %"
```

```
831  PRINT ""
835  PRINT "DO YOU WANT ANOTHER RUN?(Y/N)"
840  GET Q$
850  IF Q$ = "Y" THEN  GOTO 157
859  PRINT : PRINT "END OF RUN - THANK YOU"
860  END
870  REM
1000  REM
1001  REM  * SUBROUTINE FOR CHANGING PARAMETERS
1002  REM
1010  PRINT : PRINT "THE PHYSICAL DIMENSIONS OF THE PUMP
      ARE:"
1020  PRINT : PRINT "PISTON DIAMETER=";DP" METRES"
1030  PRINT : PRINT "PISTON LENGTH=";PSTL" METRES"
1040  PRINT : PRINT "NO. OF HOLES IN PISTON=";NH
1050  PRINT : PRINT "DIAMETER OF EACH HOLE=";DR" METRES"
1060  PRINT : PRINT "OUTER VALVE DIAMETER=";VO" METRES"
1070  PRINT : PRINT "INNER VALVE DIAMETER=";VI" METRES"
1080  PRINT : PRINT "VALVE GAP=";VGAP" METRES"
1090  PRINT : PRINT "VALVE MASS=";AMAS" KG"
1095  PRINT : PRINT "SPECIFIC DENSITY OF VALVE FLAP=";SG
1100  PRINT : PRINT "CHANGE THE PHYSICAL DIMENSIONS?
      (Y/N)"
1110  GET Q$
1111  HOME
1120  IF Q$ = "N" THEN  GOTO 1240
1130  PRINT : PRINT "KEY IN THE FOLLOWING IN METRES"
1140  PRINT : INPUT "PISTON DIAMETER=";DP
1160  PRINT : INPUT "PISTON LENGTH=";PSTL
1170  PRINT : INPUT "NO.OF HOLES IN PISTON=";NH
1180  PRINT : INPUT "DIAMETER OF EACH HOLE=";DR
1190  PRINT : INPUT "OUTER VALVE DIAMETER=";VO
1210  PRINT : INPUT "INNER VALVE DIAMETER=";VI
1220  PRINT : INPUT "VALVE GAP=";VGAP
1230  PRINT : INPUT "VALVE MASS IN KG=";AMAS
1233  PRINT : PRINT "SPECIFIC DENSITY OF PVC = 1.18"
1234  PRINT : PRINT "SPECIFIC DENSITY OF BRASS = 8.74"
1235  PRINT : INPUT "SPECIFIC DENSITY OF VALVE FLAP=";SG
1240  PRINT : PRINT "THE EMPIRICAL CONSTANTS ARE:-"
1250  PRINT : PRINT "PISTON RING LEAKAGE CONSTANT=";CLEAK
1260  PRINT : PRINT "FRICTION FORCE=";FF" NEWTONS"
1270  PRINT : PRINT "CHANGE THE EMPIRICAL CONSTANTS? (Y/N)"
1280  GET Q$
1281  HOME
1290  IF Q$ = "N" THEN  GOTO 1311
1300  PRINT : INPUT "PISTON RING LEAKAGE CONSTANT=";CLEAK
1310  PRINT : INPUT "FRICTION FORCE IN NEWTONS=";FF
```

```
1311  PRINT : PRINT "DEPTH OF WELL=";DEPTH" METRES"
1312  PRINT : PRINT "STROKE LENGTH=";STR" METRES"
1313  PRINT : PRINT "SPEED OF APPLICATION="RPM" STR/MIN"
1314  PRINT : PRINT "CHANGE THE PARAMETERS? (Y/N)"
1315  GET QS
1316  IF QS = "N" THEN  GOTO 1360
1320  PRINT : PRINT "KEY IN THE FOLLOWING:-"
1330  PRINT : INPUT "DEPTH OF WELL IN METRES=";DEPTH
1340  PRINT : INPUT "STROKE LENGTH IN METRES=";STR
1350  PRINT : INPUT "SPEED OF APPLICATION IN STR/MIN=";RPM
1360  PRINT : PRINT "THANK YOU - PLEASE WAIT"
1370  RETURN
```

Sample Output of Typical Run Using Program TMODEL

```
]RUN TMODEL

THE PHYSICAL DIMENSIONS OF THE PUMP ARE:

PISTON DIAMETER=.054 METRES

PISTON LENGTH=.0762 METRES

NO. OF HOLES IN PISTON=6

DIAMETER OF EACH HOLE=9E-03 METRES

OUTER VALVE DIAMETER=.04 METRES

INNER VALVE DIAMETER=.011 METRES

VALVE GAP=2.87E-03 METRES

VALVE MASS=.0135 KG

SPECIFIC DENSITY OF VALVE FLAP=1.18

CHANGE THE PHYSICAL DIMENSIONS? (Y/N)   N

THE EMPIRICAL CONSTANTS ARE:-

PISTON RING LEAKAGE CONSTANT=3.08E-03

FRICTION FORCE=15.6 NEWTONS

CHANGE THE EMPIRICAL CONSTANTS? (Y/N)   N
```

```
DEPTH OF WELL=5.84 METRES

STROKE LENGTH=.0762 METRES

SPEED OF APPLICATION=50 STR/MIN

CHANGE THE PARAMETERS? (Y/N)            N

THANK YOU - PLEASE WAIT

THE TOTAL VALVE DELAY IS .0297082728

THE VOLUMETRIC EFFICIENCY IS 85.9 %

AND MECHANICAL EFFICIENCY IS 73.32 %

DO YOU WANT ANOTHER RUN?(Y/N)           N

END OF RUN - THANK YOU

]
```

APPENDIX B

Interactive Computer Program for Automatic Acquisition and Processing of Data from Handpump Testing Rig

The experimental set up (as shown in Fig. 10) consists of the following components

- Proof ring with four dynamic strain gauges (Tokyo Sokki strain gauges, type FLA-10-11)

- Displacement transducer (Radio Spares conductive plastic servo potontiometers, type 173-596)

- Dynamic strainmeter (Tokyo Sokki, type DT-6A)

- Digitizing storage oscilloscope (Tektronix Model 5223 with general purpose interphase bus [GPIB])

- Microcomputer (Apple II-compatible with GPIB card)

The computer program[1] will run with other makes of the first three items but, if an oscilloscope other than the Tektronix 5223 is used, statements 3980 to 4340 should be changed to suit the particular instrument.

Additional Programing Instructions

Because of the length of the program filename LABTEST, it should be compiled and the compiled version LABTEST.OBJ should be used. This reduces running time as well as improves the efficiency of computer memory use.

---

[1]The program is available from the author on floppy disk.

The following additional standard subroutines are required

- TASC, PASS0, PASS1, and PASS2 for compilation of APPLESOFT subroutines;
- RUNTIME for running a compiled program; and
- NEWSHAPES, a graphic subroutine.

To compile an Applesoft program filename LABTEST to obtain LABTEST.OBJ, type the statement RUN TASC (CR), which calls and uses the subroutines PASS0, PASS1, and PASS2. (CR) refers to depressing the "carriage return" key.

An example of such a compilation is

```
]RUN TASC  (CR)

MICROSOFT TASC
V 2.0, 10/7/81
COPYRIGHT  (C)
1981 MICROSOFT

SOURCE FILE? LABTEST (CR)


OBJECT CODE FILE:
(DEFAULT LABTEST.OBJ)? (CR)


MEMORY USAGE:
DEFAULT CONFIGURATION? N (CR)

ALTERNATE CONFIGURATION:

ADDRESS FOR LIBRARY:
(DEFAULT 2051)?  (CR)

ADDRESS FOR PROGRAM:
(NUMBER, 'HGR1', 'HGR2',
OR DEFAULT END OF LIBRARY)? HGR1  (CR)

ADDRESS FOR VARIABLES:
(DEFAULT END OF PROGRAM)?  (CR)
```

```
LIBRARY OCCUPIES 2051 - 6063
PROGRAM BEGINS AT 16384
VARIABLES BEGIN AT END OF PROGRAM

ARE THESE ADDRESSES CORRECT? Y (CR)


OPTIONS:
DEFAULT CONFIGURATION? (CR)
```

The program will proceed to compile LABTEST and store the compiled program in LABTEST.OBJ.

To run the compiled version LABTEST.OBJ, type the following statements

```
BLOAD RUNTIME (CR)

BRUN LABTEST.OBJ (CR)
```

Listing of computer program LABTEST

```
10  REM   ** AN APPLESOFT INTERACTIVE PROGRAM FOR AUTOMATIC ACQUISITION
11  REM      AND PROCESSING OF DATA FROM AN EXPERIMENTAL RIG FOR TESTING
12  REM      THE PERFORMANCE OF RECIPROCATING WATER PUMPS **
20  REM    ! INTEGER I,J,N
30  HOME : PRINT  CHR$ (7): PRINT  CHR$ (7): PRINT "SWITCH ON OSCILLOSCOPE NOW": PRINT
40  DIM B%(1020),C%(1020),SL(5),RS(5),VEL(5,15),EFF(5,15)
44  PRINT :D$ =  CHR$ (4)
45  PRINT : PRINT D$;"BLOAD NEWSHAPES"
50  CLEAR
60 Z$ =  CHR$ (26):D$ =  CHR$ (4)
70 PP$ = D$ + "PR#1":PK$ = D$ + " PR#1":PQ$ = D$ + "PR#0"
72  PRINT : PRINT : PRINT "WHAT TYPE OF UNIT SYSTEM DO YOU WANT TO USE?": PRINT : PRINT
    "1.S.I.UNIT"
74  PRINT "2.IMPERIAL UNIT": PRINT
75  PRINT : PRINT "TYPE IN THE NUMBER OF YOUR CHOICE": GET WU: ON WU GOTO 76,77
76 LB$ = "kgf":L$ = "cm":J$ = "Nm":CH$ =  CHR$ (3) +  CHR$ (13):U = 0.0981: GCTO 78
77 LB$ = "lbf":L$ = "in":J$ = "lbf-in":CH$ =  CHR$ (34):U = 1: GOTO 78
78  PRINT : PRINT : PRINT "DO YOU WANT THE CALCULATIONS TO BE": PRINT "PRINTED OUT ON THE
    PRINTER?(Y/N)": GET F$
79  IF F$ = "N" THEN PK$ = ""
90  POKE 232,78: POKE 233,27
100  REM  ** MENU A
110  HOME
120  PRINT "MENU A": PRINT : PRINT "DO YOU WANT TO": PRINT : PRINT "1.CALIBRATE THE
     PROOF-RING"
130  PRINT "2.ACQUIRE DATA FOR EFFICIENCY": PRINT "  VERSUS SPEED"
```

```
150  PRINT "3.RETRIEVE DATA FROM FILE"
160  PRINT "4.EXIT TO MENU F"
170  PRINT : PRINT "TYPE IN THE NUMBER OF YOUR CHOICE": GET QA
180  ON QA GOTO 200,600,190,1700
185  REM  ** MENU B
190  HOME : PRINT "MENU B": PRINT : PRINT "WHICH TYPE OF DATA DO YOU": PRINT "WANT TO
     RETRIEVE?": PRINT
191  PRINT "1.STRAIN VS DISPLACEMENT PLOT"
192  PRINT "2.EFFICIENCY VS SPEED PLOT"
194  PRINT "3.EFF VS SPEED PLOT,TO BE SUPERIMPOSED"
196  PRINT "4.EFFICIENCY VS SPEED TABLE"
198  PRINT : PRINT "TYPE IN THE NUMBER OF YOUR CHOICE": GET QB
199  ON QB GOSUB 2100,2100,3000,2100: PRINT : ON QB GOTO 985,1693,1693,1693
200  REM  ** CALIBRATION OF PROOF-RING
202 Z% = 0
205  HOME : PRINT "YOU ARE NOW IN CALIBRATION MODE": PRINT
210 BSUM = 0
230  PRINT
240  PRINT "WEIGHT APPLIED TO THE PROOF-RING(";LB$;")=";: INPUT WT
245  PRINT : GOSUB 4000: PRINT
250  FOR I = 1 TO 1015
260 BSUM = BSUM + B%(I)
270  NEXT
280 Z% = Z% + 1
290 BMEAN(Z%) = BSUM / 1015
300 WT(Z%) = WT
310  PRINT PK$: PRINT "Strain          =";BMEAN(Z%);" units"
320  PRINT "Weight          =";WT;LB$: PRINT PQ$
```

```
330  PRINT : PRINT "DO YOU WANT TO ACQUIRE ANOTHER CALIBRATION POINT?(Y/N)": GET F$: PRINT
340  IF F$ = "Y" THEN  GOTO 210
350  PRINT "LINEAR REGRESSION ANALYSIS OF THE CALIBRATION....": PRINT
360 XM = 0:YM = 0
370  FOR I = 1 TO Z%
380 XM = XM + BMEAN(I):YM = YM + WT(I)
390  NEXT
400 SXX = 0:SYY = 0:XY = 0:XM = XM / Z%:YM = YM / Z%
410  FOR I = 1 TO Z%
420 SXX = SXX + (BMEAN(I) - XM) ^ 2
430 SYY = SYY + (WT(I) - YM) ^ 2
440 XY = XY + (BMEAN(I) - XM) * (WT(I) - YM)
450  NEXT
460 CCEF = XY / ( SQR (SXX * SYY))
470 SF = XY / SXX
480  PRINT FK$
490  PRINT "Correl.Coef.     =";CCEF
500  PRINT "Strain Factor    =",SF;LB$;"/unit"
510  PRINT PQ$
515  REM  ** SENDING STRAIN FACTOR VALUE TO DISK
520  PRINT : PRINT D$;"OPEN SF": PRINT D$;"WRITE SF"
530  PRINT SF
540  PRINT D$;"CLOSE SF"
550  PRINT "TYPE ANY KEY TO CONTINUE": GET F$: PRINT
555  GOTO 110
560  REM  ** INPUT WORK COMPUTATION
561  HOME : PRINT "DO YOU START WITH NEW": PRINT "STROKE LENGTH/RING SIZE?(Y/N)";: GET F$:
     PRINT
```

```
562  IF FS = "Y" THEN 564
563  GOTO 1680
564 D% = D% + 1:E% = 1: GOTO 625
600 E% = E% + 1:D% = 1: GOTO 620
620  PRINT "WATER HEAD(";L$;")=";: INPUT HT
625  PRINT "RING THICKNESS(";L$;")=";: INPUT RS(D%)
630  PRINT "STROKE LENGTH(";L$;")=";: INPUT SL(D%): GOTO 640
635 E% = E% + 1:D% = D%
640  INPUT "SPEED IN STROKE/MIN =";VEL(D%,E%)
650  REM
660  GOSUB 3990
670  REM
680 ZZ = 0
690  FOR I = 2 TO 1016
700 H% = C%(I) - C%(I - 1): IF  ABS (H%) > 4 THEN C%(I) =  SGN (H%) * 4 + C%(I - 1):ZZ =
    ZZ + 1
710  NEXT
715  PRINT "NO.OF SIGNALS CUT-OFF=";ZZ
720  IF ZZ < 250 THEN 729
721 ZZ = 0: PRINT "ACQUIRED DATA BAD": PRINT "PLEASE REACQUIRE"
723  PRINT : PRINT "HIT ANY KEY WHEN READY";: GET FS: PRINT : GOTO 660
729  PRINT "NORMAL RANGE IS FROM 150 TO 250": PRINT
730 ZZ = 0: PRINT "TYPE ANY KEY TO CONTINUE"
740  GET RS
750  FOR I = 400 TO 1015
760 E1% =  ABS (C%(I) - C%(1)):E2% =  ABS (B%(I) - B%(1))
770  IF E1% < 8 AND E2% < 8 THEN 800
780  NEXT
```

```
790  PRINT "INCOMPLETE LOOP": PRINT "PLEASE REACQUIRE": PRINT "HIT ANY KEY WHEN READY";: GET
     F$: PRINT : GOTO 660
800 PERIOD% = I
810  FOR I = PERIOD% TO 1015
820 E3% =  ABS (C%(I + 1) - C%(1)):E4% =  ABS (B%(I + 1) - B%(1))
830  IF E3% < E1% AND E4% < E2% THEN E1% = E3%:E2% = E4%:PERIOD% = I: GOTO 850
840  GOTO 860
850  NEXT
860  PRINT "Period ";PERIOD%
870 CB =  - 1E38:CS = 1E38:BB =  - 1E38:BS = 1E38
880  FOR I = 1 TO PERIOD%
890  IF BB < B%(I) THEN BB = B%(I)
900  IF BS > B%(I) THEN BS = B%(I)
910  IF CB < C%(I) THEN CB = C%(I)
920  IF CS > C%(I) THEN CS = C%(I)
930  NEXT
940 CF = 264 / (CB - CS)
950 BF = 165 / (BB - BS)
960  REM
970  GOSUB 5000: GOTO 985
980  REM  ** MENU C
985  HOME : PRINT "MENU C": PRINT : PRINT "DO YOU WANT TO ": PRINT : PRINT "1.CONTINUE WITH
     CALCULATIONS"
990  PRINT "2.SEND STRAIN DISPL.DATA TO FILE"
1000  PRINT "3.PLOT THE STRAIN-DISPL.LOOP"
1020  PRINT "4.IGNORE ACQUIRED DATA AND TO": PRINT "  REACQUIRE"
1025  PRINT "5.EXIT TO MENU F"
1030  PRINT : PRINT "TYPE IN THE NUMBER OF YOUR CHOICE": GET QC
```

```
1040  ON QC GOTO 1090,1050,1060,660,1700
1050  GOSUB 8152: HOME : GOTO 985
1060  GOSUB 5000: HOME : GOTO 985
1070  REM
1080  REM  ** CALCULATION OF FORCE VS DISPL.LOOP AREA
1090  PRINT : PRINT "WAIT": PRINT "CALCULATING FORCE VS DISP.LOOP AREA"
1100  REM
1110 A = (B%(1) + B%(PERIOD%)) / 2 * (C%(1) - C%(PERIOD%))
1120  FOR I = 1 TO PERIOD% - 1
1130 A = A + (B%(I + 1) + B%(I)) / 2 * (C%(I + 1) - C%(I))
1140  NEXT
1150  PRINT : PRINT D$;"OPEN SF": PRINT D$;"READ SF"
1160  INPUT S$:SF =  VAL (S$)
1170  PRINT D$;"CLOSE SF"
1180  PRINT PK$
1190  PRINT "Area                =";A;" sq.units"
1200  PRINT "Strain Factor       =";SF;LB$;"/unit"
1210  PRINT "Stroke Length       =";SL(D%);L$
1220 DF = SL(D%) / (CB - CS)
1230  PRINT "Displ. Factor       =";DF;L$;"/unit"
1240  PRINT "Work Input          =";A * SF * DF * U;J$: PRINT : PRINT
1250  PRINT PQ$
1260 Y% = Y% + 1
1270 WINP = WINP + A * DF * SF * U
1280  PRINT "DO YOU WISH TO REACQUIRE TO GET AN AVERAGE READING?(Y/N)";: GET F$: PRINT
1290  IF F$ = "Y" THEN  GOTO 660
1300 WINP = WINP / Y%
1310  PRINT PK$: PRINT "Ave. Work Input  =";WINP;J$: PRINT PQ$
```

```
1315  REM  ** OUTPUT WORK COMPUTATION
1320  PRINT : PRINT : PRINT "OUTPUT WORK COMPUTATION": PRINT
1330  PRINT
1340  PRINT "Measure Weight of water collected for a fixed number of strokes"
1350  PRINT : PRINT
1360  PRINT "WEIGHT OF WATER COLLECTED(";LB$;")=";: INPUT WTW
1370  INPUT "NUMBER OF STROKES TAKEN =";S%
1380  PRINT : PRINT
1390 W% = W% + 1
1400  PRINT PK$: PRINT "Work Output       =";HT * WTW * U / S%;J$: PRINT PQ$
1410 WOUTPUT = WOUTPUT + HT * WTW * U / S%
1420  PRINT : PRINT "DO YOU WANT TO TAKE ANOTHER MEASUREMENT"
1425  PRINT "TO OBTAIN THE AVERAGE VALUE?(Y/N)";: GET F$: PRINT : PRINT
1430  IF F$ = "Y" THEN 1360
1440 WOUTPUT = WOUTPUT / W%
1450  PRINT PK$: PRINT "Ave. Work Output =";WOUTPUT;J$: PRINT PQ$
1460 EFF(D%,E%) = WOUTPUT * 100 / WINP: PRINT PK$
1470  PRINT "Efficiency (%)    =";EFF(D%,E%)
1480  PRINT "Speed (rev/min)   =";VEL(D%,E%): PRINT PQ$
1490  PRINT : PRINT "HIT ANY KEY TO CONTINUE": GET F$: PRINT
1510  REM
1520  GOSUB 6000
1530  REM  ** MENU D
1535  HOME : PRINT "MENU D": PRINT : PRINT "DO YOU WANT TO": PRINT : PRINT "1.CONTINUE WITH
      ANOTHER READING"
1540  PRINT "2.SEND EFF VS SPEED DATA TO FILE"
1550  PRINT "3.PLOT EFF VS SPEED CURVE"
1560  PRINT "4.PRINT OUT THE EFF VS SPEED TABLE"
```

```
1580  PRINT "5.EXIT TO MENU F"
1590  PRINT : PRINT "TYPE IN THE NUMBER OF YOUR CHOICE": GET QD
1600  ON QD GOTO 1640,1610,1620,1630,1650
1610  GOSUB 8040: GOTO 1535
1620  GOSUB 6000: GOTO 1535
1630  GOSUB 7000: GOTO 1535
1640 WINP = 0:WOUTPUT = 0:Y% = 0:W% = 0: GOTO 561
1650  GOTO 1700
1660  REM
1670  REM
1680  PRINT : PRINT "TO ACQUIRE AT NEXT SPEED?(Y/N)"
1690  GET Q$: IF Q$ = "Y" THEN 635
1691  IF Q$ = "N" THEN 660
1692  REM  ** MENU E
1693  HOME : PRINT "MENU E": PRINT : PRINT "DO YOU WANT TO"
1694  PRINT : PRINT "1.CONTINUE TEST WITH RETRIEVED": PRINT "DATA"
1695  PRINT "2.START FRESH TEST OR TO RETRIEVE": PRINT "  NEW DATA FROM FILE"
1696  PRINT "3.HAVE MORE COPIES OF THE RETRIEVED DATA"
1697  PRINT "4.EXIT FROM PROGRAM"
1698  PRINT : PRINT "TYPE IN THE NUMBER OF YOUR CHOICE"
1699  GET QE: ON QE GOTO 561,50,1535,1770
1700  REM  ** MENU F
1705  HOME : PRINT "MENU F": PRINT : PRINT "DO YOU WANT TO": PRINT : PRINT "1.RECALIBRATION
      OF PROOF-RING"
1710  PRINT "2.START FRESH TEST OR TO RETRIEVE": PRINT "  DATA FROM FILE"
1730  PRINT "3.EXIT FROM PROGRAM"
1740  PRINT : PRINT "TYPE IN THE NUMBER OF YOUR CHOICE"
1750 GET QF: ON QF GOTO 1755,50,1770
```

```
1755 WINP = 0:WOUTPUT = 0:Y% = 0:W% = 0: GOTO 200
1760  REM
1770  PRINT : PRINT D$;"CATALOG": END
2000  REM  ** SUBROUTINE FOR RETRIEVE STRAIN VS DISPL.OR EFF VS SPEED DATA FROM DISK
2100  PRINT : INPUT "FILENAME FOR DATA =";S$: PRINT
2110  ONERR  GOTO 2270
2120  PRINT D$;"OPEN";S$: PRINT D$;"READ";S$
2130  INPUT A$: PRINT A$:
2140  ON QB GOTO 2150,2210,2280,2210
2145  REM  ** RETRIEVE STRAIN VS DISPL.DATA
2150  INPUT A$,B$,C$,E$:HT =  VAL (A$):SL(1) =  VAL (B$):RS(1) =  VAL (C$):VEL(1,1) =
      VAL (E$)
2160  INPUT A$,B$,C$,E$:PERIOD% =  VAL (A$):CS =  VAL (B$):CB =  VAL (C$):CF =  VAL (E$)
2165  INPUT A$,B$:BS =  VAL (A$):BF =  VAL (B$)
2170  FOR I = 1 TO 1015
2180  INPUT A$,B$:B%(I) =  VAL (A$):C%(I) =  VAL (B$)
2190  NEXT
2195 D% = 1:E% = 1
2200  GOTO 2270
2205  REM  ** RETRIEVE EFF VS SPEED DATA
2210  INPUT D%,E%,HT
2220  FOR I = 1 TO D%: INPUT SL(I): INPUT RS(I)
2230  FOR J = 1 TO 15: INPUT EFF(I,J),VEL(I,J)
2240  NEXT : NEXT
2250  REM
2260  REM
2270  POKE 216,0: PRINT D$;"CLOSE";S$:FR =  FRE (0)
2280  ON QB GOSUB 5000,6000,3000,7000
```

```
2290  RETURN
3000  REM   ** SUBROUTINE TO RETRIEVE EFF VS SPEED DATA FROM DISK AND SUPERIMPOSE THEM
3005 IP = 0
3010  PRINT : PRINT "GIVE NAME OF FILE CONTAINING"
3020  PRINT "THE EFF VS SPEED CURVE": PRINT
3030  INPUT F$
3040  PRINT D$;"OPEN";F$
3050  PRINT D$;"READ";F$
3055  INPUT A$
3060  INPUT D%: INPUT E%: INPUT HT
3065  FOR K = 1 TO D%:I = K + IP: INPUT SL(I): INPUT RS(I)
3070  FOR J = 1 TO 15
3080  INPUT EFF(I,J): INPUT VEL(I,J)
3090  NEXT : NEXT
3100  PRINT D$;"CLOSE";F$
3110  HOME : PRINT "DO YOU WANT TO SUPERIMPOSE MORE CURVE?(Y/N)"
3120  GET Q$
3130  IF Q$ = "Y" THEN IP = I:FR =  FRE (0): GOTO 3010
3140 D% = I
3150  GOSUB 6000
3160  RETURN
3170  REM
3980  REM  ** SUBROUTINE FOR DATA ACQUISITION FROM TEKTRONIX 5223
3990  HOME : PRINT "PRESS THE (L VS R) BUTTON OF THE SCOPE"
3995  PRINT "AND SAVE THE LOOP WHEN YOU ARE": PRINT "SATISFIED WITH IT": PRINT
4000  PRINT "HIT ANY KEY TO COMMENCE ACQUISITION";: GET F$: PRINT
4010 A$ = "L"
4020  PRINT : PRINT "ACQUISITION IN PROGRESS"
```

```
4030  GOSUB 4070
4040  PRINT : PRINT "DATA UNDER PROCESS"
4050  RETURN
4060  REM
4070  PRINT D$;"PR#3": PRINT D$;"IN#3"
4080  PRINT "RA"
4090  PRINT "WT!";Z$;"ACCESS ";A$
4100  PRINT "WT!";Z$;"ASCII;CURVE?"
4110  PRINT "LF1"
4120  PRINT "RDA";Z$;: GOSUB 4200
4130  IF N$ <  > "A" AND A$ = "R" THEN 4080
4140  PRINT "UT"
4150  PRINT "LA"
4160  PRINT D$;"PR#0": PRINT D$;"IN#0"
4170  RETURN
4180  REM
4190  REM
4200  FOR I = 1 TO 6
4210  GET C$: PRINT C$;
4220  NEXT
4230  FOR I = 1 TO 1015:B$ = ""
4240  GET C$: PRINT C$;
4250  IF A$ = "R" THEN 4270
4260  IF C$ = "," THEN B%(I) =  VAL (B$): GOTO 4290
4270  IF C$ = "," THEN C%(I) =  VAL (B$): GOTO 4290
4280 B$ = B$ + C$: GOTO 4240
4290  NEXT
4300 B$ = "":C$ = ""
```

```
4310  GET C$: PRINT C$;:B$ = B$ + C$
4320  IF A$ = "R" THEN 4340
4330  IF  ASC (C$) = 13 THEN B%(1016) =  VAL (B$):A$ = "R": RETURN
4340  IF  ASC (C$) = 13 THEN C%(1016) =  VAL (B$):A$ = "L": RETURN
4350  GOTO 4310
5000  REM  ** SUBROUTINE FOR PLOTTING STRAIN VS DISPL.GRAPH
5005  HGR : HCOLOR= 3: ROT= 0: SCALE= 1
5010  POKE  - 16302,0
5020  HPLOT 12,0 TO 12,165 TO 279,165
5030 C0 = (C%(1) - CS) * CF + 15:B0 = 165 - (B%(1) - BS) * BF
5040 CZ = C0:BZ = B0
5050  FOR I = 1 TO PERIOD%
5060 XX = (C%(I) - CS) * CF + 15:YY = 165 - (B%(I) - BS) * BF
5070  HPLOT C0,B0 TO XX,YY
5080 C0 = XX:B0 = YY
5090  NEXT
5100  HPLOT C0,B0 TO CZ,BZ
5110 M$ = "NIARTS":X% = 2: ROT= 48: GOSUB 9170: ROT= 0
5120 M$ = "DISPLACEMENT":X% = 1:X1% = 80:Y1% = 185: GOSUB 9170
5130 Y1% = 170
5140  GCSUB 9250
5150  TEXT : HOME
5160  RETURN
5170  REM
6000  REM  ** SUBROUTINE FOR PLOTTING EFF VS SPEED CURVE
6005  HGR : HCOLOR= 3: SCALE= 1: ROT= 0
6010  POKE  - 16302,0
6020  GOSUB 9000: GCSUB 9120
```

```
6030  FOR I = 1 TO D%
6040  FOR J = 1 TO 15
6050  IF EFF(I,J) = 0 AND VEL(I,J) = 0 THEN J = 15: GOTO 6090
6060  IF EFF(I,J) <  = 35 OR EFF(I,J) > 100 THEN 6090
6070 V1 = VEL(I,J) * 4 - 10:EF1 = 249 - EFF(I,J) * 2.5
6080  DRAW I + 64 AT V1,EF1
6090  NEXT
6100  DRAW I + 64 AT 175,165 - 8 * I
6103 M$ = "RS=" +  STR$ (RS(I)) + " SL=" +  STR$ (SL(I)) + CH$
6105 X1% = 175:Y1% = 162 - 8 * I: GOSUB 9170
6110 Y1% = 145
6120  GOSUB 9250
6130  NEXT
6140  TEXT
6150  RETURN
7000  REM  ** SUBROUTINE FOR PLOTTING EFF VS SPEED TABLE
7005  INPUT "Date ? ";B$:C$ = "Date: " + B$
7010  PRINT "To Printer? (Y/N)";: GET F$: PRINT
7020  IF F$ = "Y" THEN  PRINT PP$
7030  PRINT C$
7040  PRINT "Stroke  Ring","Efficiency"; TAB( 30);"Speed"
7045  PRINT "(";L$;")";: HTAB 9: PRINT "(";L$;")"
7050  FOR I = 1 TO D%: PRINT SL(I);: HTAB 9: PRINT RS(I)
7060  FOR J = 1 TO 15
7070  IF EFF(I,J) = 0 AND VEL(I,J) = 0 THEN J = 15: GOTO 7090
7075  IF EFF(I,J) <  = 20 OR EFF(I,J) > 100 THEN 7090
7080  PRINT ,EFF(I,J); TAB( 30);VEL(I,J)
7090  NEXT : NEXT
```

```
7100  PRINT PQ$
7110  PRINT "Hit any key for MENU": GET F$
7120  RETURN
8030  REM  ** SUBROUTINE FOR SENDING EFF VS SPEED DATA TO DISK
8040  HOME : INPUT " Filename ?";S$
8050  INPUT "Remarks :";F$
8060  PRINT D$;"OPEN";S$: PRINT D$;"WRITE";S$
8070  PRINT F$
8090  PRINT D%: PRINT E%: PRINT HT
8100  FOR I = 1 TO D%
8110  PRINT SL(I): PRINT RS(I)
8120  FOR J = 1 TO 15
8130  PRINT EFF(I,J): PRINT VEL(I,J)
8140  NEXT : NEXT
8150  GOTO 8210
8151  REM  ** SUBROUTINE FOR SENDING STRAIN VS DISPL.DATA TO DISK
8152  HOME : INPUT "Filename ?";S$
8154  INPUT "Remark :";F$
8156  PRINT D$;"OPEN";S$: PRINT D$;"WRITE";S$
8158  PRINT F$
8160  PRINT HT: PRINT SL(D%): PRINT RS(D%): PRINT VEL(D%,E%)
8170  PRINT PERIOD%: PRINT CS: PRINT CB: PRINT CF: PRINT BS: PRINT BF
8180  FOR I = 1 TO 1015
8190  PRINT B%(I): PRINT C%(I)
8200  NEXT
8210  PRINT D$;"CLOSE";S$
8220  RETURN
9000  REM   ** SUBROUTINE FOR PLOTTING EFF VS SPEED AXES
```

```
9005  HPLOT 30,20 TO 30,169 TO 279,169
9010  FOR I = 4 TO 9
9020 M1% = 10 * I:M$ =  STR$ (M1%):Y1% = 246 - 2.5 * M1%
9030  DRAW 45 AT 28,Y1%
9040 X% = 1:X1% = 6: GOSUB 9170
9050  NEXT
9060  FOR I = 2 TO 7
9070 M1% = 10 * I:M2% = M1% * 4 - 12:M$ =  STR$ (M1%)
9080  DRAW 9 AT M2%,169
9090 X1% = M2% - 10:Y1% = 177: GOSUB 9170
9100  NEXT
9110  RETURN
9120 M$ = "YCNEICIFFE":X% = 2: ROT= 48: GOSUB 9170: ROT= 0
9130 M$ = "SPEED OF STROKE APPLICATION ":X% = 1:X1% = 80:Y1% = 185: GOSUB 9170
9140 M$ = "VARIATION OF MECHANICAL EFFICIENCY (%) WITH":X1% = 10:Y1% = 0: GOSUB 9170
9150 M$ = "SPEED OF STROKE APPLICATION (REV/MIN) ":X1% = 40:Y1% = 11: GOSUB 9170
9160  RETURN
9170 L% =  LEN (M$)
9180 FR =  FRE (0)
9190  FOR N = 1 TO L%
9200 B$ =  MID$ (M$,N,1):B% =  ASC (B$)
9210  IF B% > 64 THEN B% = B% - 64
9220  ON X% GOTO 9230,9240
9230  DRAW B% AT X1% + 6 * N,Y1%: NEXT : RETURN
9240  DRAW B% AT 3,40 + 8 * N: NEXT : RETURN
9250 M$ = "TO PRINTER? (Y/N)":X1% = 60
9260  GOSUB 9170: HCOLOR= 0: GOSUB 9170: HCOLOR= 3:KEY% =  PEEK (49152): POKE 49168,0
9270  IF KEY% < 128 THEN 9260
```

```
9280  IF KEY% = 217 THEN  GOSUB 9300
9290  RETURN
9300  REM  ** SUBROUTINE FOR CHOICE OF SIZE OF PRINT OUT
9305  TEXT : HOME
9310  VTAB 7: HTAB 4
9320  PRINT " Choose Size of Print Out"
9330  HTAB 10: PRINT "1. One To One"
9340  HTAB 10: PRINT "2. Enlarged"
9350  HTAB 10: GET F$: PRINT
9360  IF F$ = "1" THEN  POKE 1913,1: POKE 1145,75
9370  IF F$ = "2" THEN  POKE 1913,65: POKE 1145,76
9380  PRINT PP$: PRINT  CHR$ (17): PRINT PQ$
9390  POKE  - 16304,0: POKE  - 16297,0
9400  RETURN
9410  REM
```

Sample Results of Typical Running of LABTEST

```
]BLOAD RUNTIME  (CR)
]BRUN LABTEST.OBJ  (CR)

SWITCH ON OSCILLOSCOPE NOW


WHAT TYPE OF UNIT SYSTEM DO YOU WANT TO USE?

1.  S.I. UNIT
2.  IMPERIAL UNIT


TYPE IN THE NUMBER OF YOUR CHOICE  1 (CR)


DO YOU WANT THE CALCULATIONS TO BE
PRINTED OUT ON THE PRINTER?(Y/N)    Y  (CR)

MENU A

DO YOU WANT TO

1.  CALIBRATE THE PROOF-RING
2.  ACQUIRE DATA FOR EFFICIENCY VERSUS SPEED
3.  RETRIEVE DATA FROM FILE
4.  EXIT TO MENU F


TYPE IN THE NUMBER OF YOUR CHOICE  1 (CR)

YOU ARE NOW IN CALIBRATION MODE


WEIGHT APPLIED TO THE PROOF-RING(kgf)=?0 (CR)

HIT ANY KEY TO COMMENCE ACQUISITION (CR)

ACQUISITION IN PROGRESS
Strain          = -261.4944581 units
Weight          = 0 kgf

DO YOU WANT TO ACQUIRE ANOTHER CALIBRATION POINT?(Y/N)
   Y  (CR)

WEIGHT APPLIED TO THE PROOF-RING(kgf)=?6.14  (CR)
```

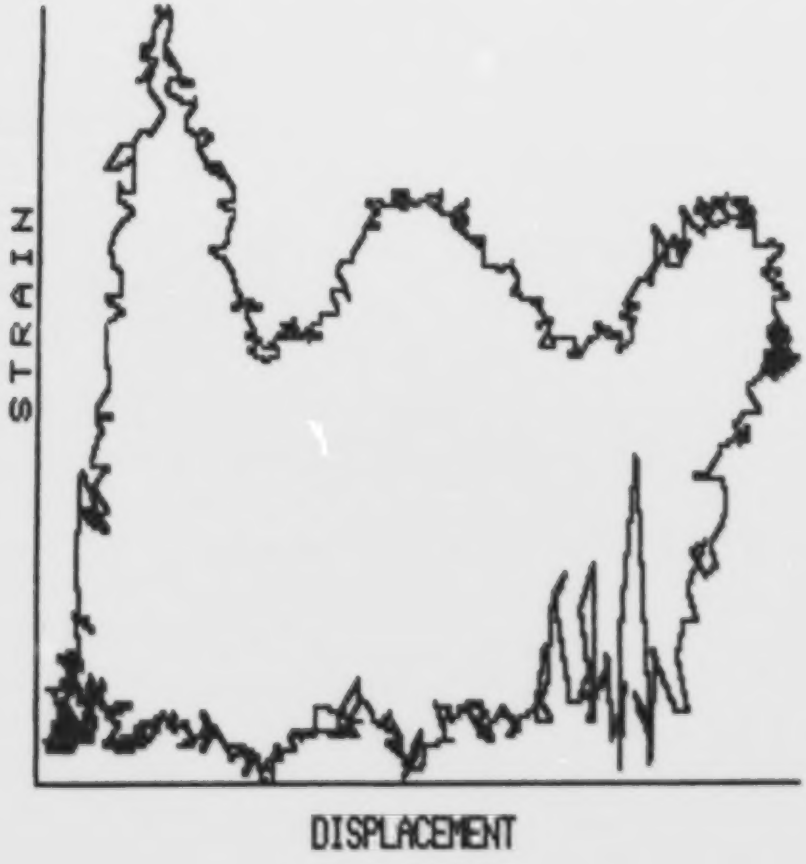
```
HIT ANY KEY TO COMMENCE ACQUISITION (CR)

ACQUISITION IN PROGRESS
Strain          = -233.7931044 units
Weight          = 6.14 kgf

DO YOU WANT TO ACQUIRE ANOTHER CALIBRATION POINT?(Y/N)
   Y (CR)

WEIGHT APPLIED TO THE PROOF-RING(kgf)=? 28.9 (CR)


HIT ANY KEY TO COMMENCE ACQUISITION (CR)

ACQUISITION IN PROGRESS
Strain          = -147.748769 units
Weight          = 28.9 kgf


DO YOU WANT TO ACQUIRE ANOTHER CALIBRATION POINT?(Y/N)
   N (CR)

LINEAR REGRESSION ANALYSIS OF THE CALIBRATION ...
Correl. Coef.       = .99946653
Strain Factor       = .256557964 kgf/unit


TYPE ANY KEY TO CONTINUE (CR)


MENU A

DO YOU WANT TO

1.  CALIBRATE THE PROOF-RING
2.  ACQUIRE DATA FOR EFFICIENCY VERSUS SPEED
3.  RETRIEVE DATA FROM FILE
4.  EXIT TO MENU F


TYPE IN THE NUMBER OF YOUR CHOICE 2 (CR)
WATER HEAD (cm)=?612 (CR)
RING THICKNESS(cm)=?0.159 (CR)
STROKE LENGTH (cm)=?7.6 (CR)
SPEED IN STROKE/MIN =59 (CR)
PRESS THE (L VS R) BUTTON OF THE SCOPE
AND SAVE THE LOOP WHEN YOU ARE
SATISFIED WITH IT
```

HIT ANY KEY TO COMMENCE ACQUISITION (CR)

ACQUISITION IN PROGRESS
NO. OF SIGNALS CUT-OFF = 214
NORMAL RANGE IS FROM 150 TO 250

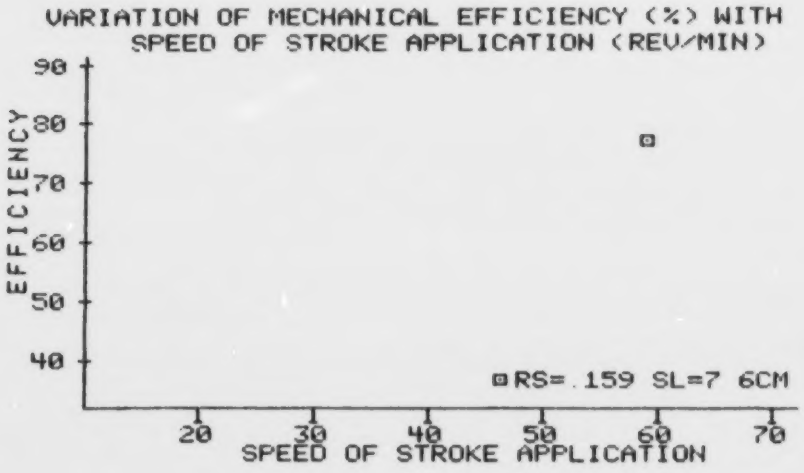
TYPE ANY KEY TO CONTINUE (CR)
Period 716

Choose Size of Print Out
1. One To One
2. Enlarged 2 (CR)



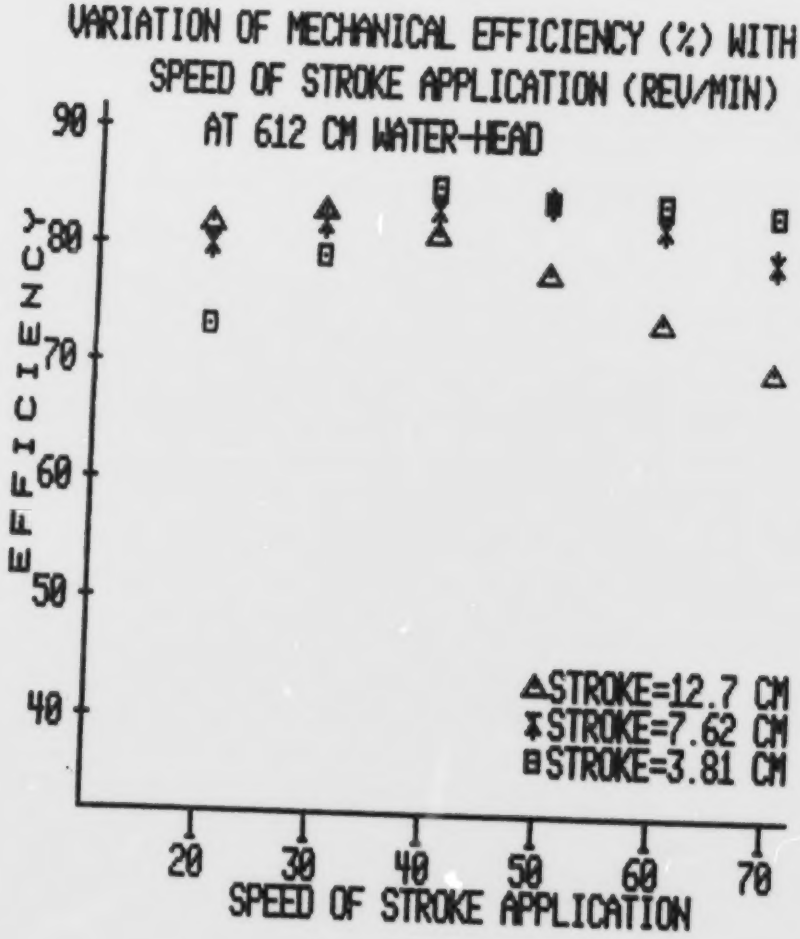
```
MENU C

DO YOU WANT TO

1.  CONTINUE WITH CALCULATIONS
2.  SEND STRAIN DISPL. DATA TO FILE
3.  PLOT THE STRAIN-DISPL. LOOP
4.  IGNORE ACQUIRED DATA AND TO REACQUIRE
5.  EXIT TO MENU F

TYPE IN THE NUMBER OF YOUR CHOICE 1 (CR)

WAIT
CALCULATING FORCE VS DISP. LOOP AREA

Area                = 25932 sq. units
Strain Factor       = .256557964 kgf/unit
Stroke Length       = 7.6 cm
Displ. Factor       = .0360189573 cm/unit
Work Input          = 23.5083235 Nm

DO YOU WISH TO REACQUIRE TO GET AN AVERAGE READING?(Y/N)
   N (CR)

Ave. Work Input     = 23.5083235 Nm


OUTPUT WORK COMPUTATION


Measure Weight of water collected for a fixed number of
   strokes

WEIGHT OF WATER COLLECTED (kgf) =?18.1 (CR)
NUMBER OF STROKES TAKEN =60 (CR)


Work Output         = 18.111222 Nm


DO YOU WANT TO TAKE ANOTHER MEASUREMENT
TO OBTAIN THE AVERAGE VALUE? (Y/N)        N (CR)


Ave. Work Output    = 18.111222 Nm
Efficiency (%)      = 77.0417424
Speed (rev/min)     = 59
```

```
HIT ANY KEY TO CONTINUE   (CR)

Choose Size of Print Out
1.  One To One
2.  Enlarged                    1   (CR)
```




```
MENU D

DO YOU WANT TO

1.  CONTINUE WITH ANOTHER READING
2.  SEND EFF VS SPEED DATA TO FILE
3.  PLOT EFF VS SPEED CURVE
4.  PRINT OUT THE EFF VS SPEED TABLE
5.  EXIT TO MENU F

TYPE IN THE NUMBER OF YOUR CHOICE   5   (CR)

MENU F

DO YOU WANT TO

1.  RECALIBRATION OF PROOF-RING
2.  START FRESH TEST OR TO RETRIEVE DATA FROM FILE
3.  EXIT FROM PROGRAM

TYPE IN THE NUMBER OF YOUR CHOICE   3   (CR)
```

Sample of Output for Complete Set of Experiments



APPENDIX C

Field Monitoring Data Collection Sheets

FORM A : WELL AND PUMP SPECIFICATIONS

Part I **Date of Entry:** _____

WELL

1. Identification number : .....
2. Geographical location : .....
3. Type: Dug/Drilled/Driven : .....
4. Diameter of well/casing : .....
5. Casing material : .....
6. Special features (filtering system/screens etc) : .....
7. Depth of well : .....
8. Depth of water table below ground level : .....
9. Date of construction of well or estimated age : .....
10. Uses of water - Washing/ Bathing/Cooking/Drinking : .....
11. Water characteristics : .....

    (a) Chemical:

        (i) ph value : .....

        (ii) Hardness : .....

        (iii) Other properties : .....

(b) Physical

(i) Colour : .....

(ii) Taste : .....

(iii) Odour : .....

(iv) Other observations:

(c) Microbiological : .....

12. Aquifer characteristics : .....

13. Distance to nearest and furthest household using well: .....

14. Estimated household served : .....

15. Estimated population served : .....

16. Remarks (if any) : .....

PART II: PUMP

1. Identification number : .....

2. Type name : .....

3. Country of manufacture : .....

4. Cylinder diameter : .....

5. Average piston stroke : .....

6. Average operating speed : .....

7. Average delivery rate : .....

8. Volumetric efficiency : .....

9. Mechanical efficiency : .....

10. Mechanical advantage of leverage system : .....

11. Monitoring attachments : .....

12. Depth of drop pipe (below ground level) : .....

13. Remarks (if any) : .....

FORM B : SITE VISIT DATA SHEET

Date of Entry : .....

Pump Indentification No. : .....

B.1 MECHANICAL PERFORMANCE

1. **Cumulative Piston Displacement:-**

   1.1 **Present counter reading : .....**

   Previous counter reading: .....

   Cumulative change : .....

   since previous visit on .....(date)

   1.2 **Counter roller diameter: (i) ..... (ii) .....**

   Average : .....

   1.3 **Counter calibration constant: .....**

   1.4 **Cumulative piston displacement: .....**

   for ..... days.

   Average piston displacement per day: .....

   1.5 **Remarks : .....**

   .....

   .....

   .....

2. **Static Water Table Level: .....**

3. **Volumetric Efficiency:- (at standard piston stroke and frequency)**

   3.1 **Theoretical delivery/standard storke: .....**

   3.2 **Standard volume used: .....**

   3.3 **Theoretical number of standard strokes required to fill standard volume: .....**

3.4 Actual number of strokes required to fill standard volume: (i) ..... (ii) .....

(ii) ..... Average : .....

3.5 Volumetric efficiency: .....

4. Check Readings:-

4.1 Counter reading at start (i).....(ii).....(iii).....

at end (i).....(ii).....(iii).....

of volumetric efficiency tests.

4.2 Average change in counter reading: .....

4.3 Average total piston travel (in volumetric test): .....

4.4 Equivalent number of standard piston stroke: .....

4.5 Volumetric efficiency: .....

5. Mechanical Efficiency:-

Attach the mechanical plotter between the upper end of the piston rod and the lever arm piston rod connector. Obtain a plot of the force-displacement diagram.

5.1 Area of Force-displacement loop (measured using a mechanical plainmeter): .....

5.2 Mechanical plotter calibration constant: .....

5.3 Mechanical efficiency: .....

6. Remarks: .....

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B.2 MAINTENANCE OPERATIONS

1. **Active maintenance time:-**

   Start time : .....

   Stop time : .....

2. **Description:** .....

   .....

   .....

   .....

   .....

   .....

3. **Dimensions of new replacement parts/Description of measure taken:** .....

   .....

   .....

   .....

   .....

   .....

4. **Personnel/Transport used:**

   4.1 **Type of vehicle:** .....

   4.2 **Estimated mileage to and from base:** .....

   4.3 **Name of Personnel** **Designation**

   ....

   ....

   ....

   ....

B.3 WEAR MEASUREMENTS

1. Wear measurements were: taken/not taken

2. Leverage Parts:-

   2.1 Top wooden bushing

   (i) Diameter at 0

   (a) .....

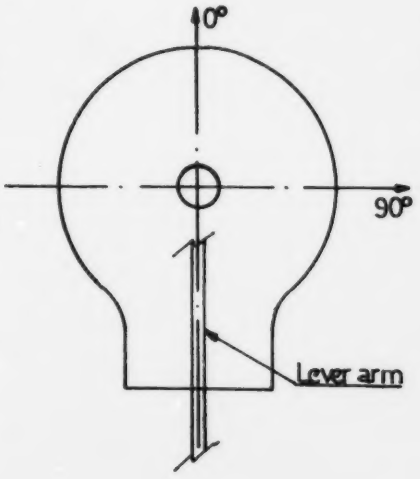
   (b) .....

   (ii) Diameter at 90

   (a) .....

   (b) .....

   Average: .....



   2.2 Piston rod

   (i) Diameter at 0

   (a).....(b).....(c).....(d).....

   (ii) Diameter at 90

   (a).....(b).....(c).....(d).....

   Average: .....

   Comments: .....

3. Piston

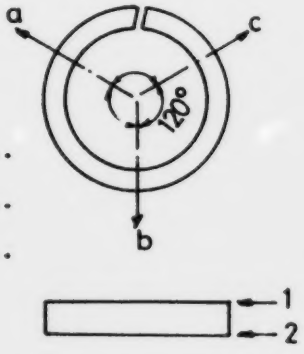
   3.1 Piston ring thickness

   (i) Top

   (a1).....(a2).....

   (b1).....(b2).....

   (c1).....(c2).....



(ii) Bottom

(a1).....(a1).....

(b1).....(b2).....

(c1).....(c2).....

Comments: .....

.....

.....

3.2 Valve Seat (depth gauge)

(a).....(b).....

(c).....(d).....

(e).....(f)..... Average: .....

3.3 Piston diameter

(i) Diameter d_1

(a)..... (b).....

(c).....

Average: .....

(ii) Diameter d_2

(a)..... (b).....

(c).....

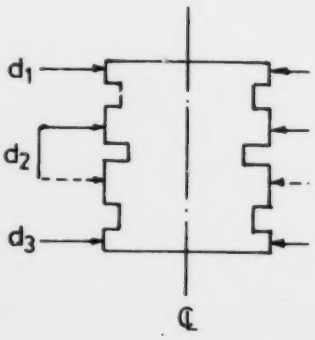
Average: .....

(iii) Diameter d_3

(a)..... (b).....

(c).....

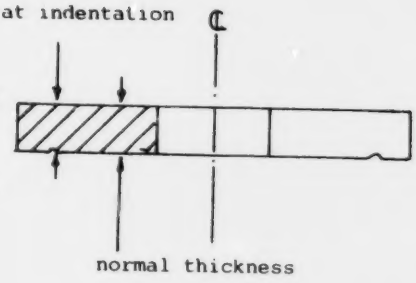
Average: .....



Comments: .....

.....

3.4 Valve plate thickness



Normal thickness

(a)..... (b).....

(c).....

Average: .....

At indentation

(a)..... (b).....

(c).....

Average: .....

Comments: .....

.....

4. Footvalve

4.1 Valve seat (depth gauge)

(a)..... (b).....

(c)..... (d).....

(e)..... (f)..... Average: .....

4.2 Valve plate thickness

Normal thickness

(a)..... (b).....

(c).....

Average: .....

At indentation

(a)..... (b).....

(c).....

Average: .....

5. Remarks on wear measurements: .....

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B.4 USER FEEDBACK

The following data was/was not obtained in this visit from a responsible user.

1. User description: Age: ..... Sex: .....

Occupation: .....

Others: .....

.....

2. Down time: .....

3. Use of water: washing/bathing/cooking/drinking

other: .....

4. Other source of water: .....

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5. Comments/Complaints/Suggestions: .....

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FORM C : FAILURES AND REPAIRS

Date of Entry : .....

Pump Identification No. : .....

1. Actual repair time :

   (a) Start time : .....

   (b) Stop time : .....

2. Down time : .....

3. Description of failures: .....

   .....

   .....

   .....

   .....

4. Description of repairs: .....

   .....

   .....

   .....

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5. Personnel/Transport Used:

   (a) Type of vehicle : .....

   (b) Estimated mileage to and from base: .....

   (c) Name of Personal Designation

   ....

   ....

   ....

FORM D : HANDPUMP CARETAKER'S DATA COLLECTION SHEET

1. Pump identification number: .....

2. Pump type: .....

3. Location : .....

4. Date of installation of pump: .....

5. Remarks: .....

.....

.....

.....

6. Counter Readings

Date and time of installation: .....

Initial reading: .....

Date	Time	Counter Reading	Remarks